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TRADEOFF STUDY FOR EXTENDED LIFE HELICOPTER TRANSMISSION

Charles W. Bowen, et al

Bell Helicopter Company

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TRADEOFF STUDY FOR EXTENDED-LIFE HELICOPTER TRANSMISSION

By

C. W. Bowen

R. D. Walker

November 1972

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-70-C-0053
BELL HELICOPTER COMPANY
A DIVISION OF BELL AEROSPACE CORPORATION
FORT WORTH, TEXAS

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This report was prepared by Bell Helicopter Company, Division of Bell Aerospace Corporation, under the terms of Contract DAAJ02-70-C-0053. It presents the results of a design trade-off study conducted to determine the operational cost impact of substantially extending the overhaul life of helicopter main transmissions.

The purpose of this contractual effort was to conduct a general trade-off study to evaluate the direct operational costs associated with increasing helicopter transmission time-between-overhaul (TBO) levels into the 3000- to 6000-hour range. This evaluation was accomplished through determination of the cost, availability, and performance for three distinct sizes of helicopter power transmission featuring twin-engine inputs of 500, 1500 and 4800 horsepower per engine.

On the basis of overhaul and transportation costs as well as fuel cost in the RVN theater, this study showed that both the 3000-hour and 6000-hour TBO design transmissions are cost effective. These increased TBO transmission designs were shown to be feasible basically through changes in component materials and through component design and integration techniques.

The technical managers for this contract were Messrs. James Gomez and Wayne A. Hudgins of the Technology Applications Division.

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November 1972

TRADEOFF STUDY FOR EXTENDED-LIFE HELICOPTER TRANSMISSION

(Bell Helicopter Report 299-099-492)

by

C. W. Bowen
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Prepared by

BELL HELICOPTER COMPANY
A Division of Bell Aerospace Corporation
Fort Worth, Texas

for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

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SUMMARY

This report presents the results of a design tradeoff study conducted to determine the operational cost impact of extending the overhaul life of the drive train components from 1200 hours to 3000 and 6000 hours on future Army helicopters.

Pertinent Army and BHC publications were reviewed to determine Time Between Overhauls (TBO) limiting factors.

An analytical review was also made of a mission profile study conducted on monitored AH-1G, UH-1H, and UH-1C helicopters in Viet Nam to determine an appropriate usage rate, power spectrum, and flight length spectrum for this study. From this review, a cubic mean power requirement of 65% takeoff power was established, and a vehicle utilization rate of 50 hours per month was justified.

The design tradeoff study was implemented through the design, l[illegible], and analysis of three power level transmissions each a[illegible]ee TBO levels. The transmissions featured twin-engine direct-drive configurations of two 500-hp, two 1500-hp, and two 4800-hp power level engines. Each transmission was initially designed for a 1200-hour TBO and was then modified as required to attain a 3000-hour TBO and again for a 6000-hour TBO.

The relative effective costs of the three transmissions as affected by TBO requirements were found to be:

	RELATIVE COST		
	1200-Hour TBO	3000-Hour TBO	6000-Hour TBO
Twin 500-HP Design	1.0	.788	.450
Twin 1500-HP Design	1.0	.875	.653
Twin 4800-HP Design	1.0	.771	.509

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13. ABSTRACT

This report presents the results of a design tradeoff study conducted to determine the operational cost impact of extending the overhaul life of the drive train components from 1200 hours to 3000 and 6000 hours on future Army helicopters.

Pertinent Army and BHC publications were reviewed to determine Time Between Overhauls (TBO) limiting factors.

An analytical review was also made of a mission profile study conducted on monitored AH-1G, UH-1H, and UH-1C helicopters in Viet Nam to determine an appropriate usage rate, power spectrum, and flight length spectrum for this study. From this review, a cubic mean power requirement of 65% takeoff power was established, and a vehicle utilization rate of 50 hours per month was justified.

The design tradeoff study was implemented through the design, layout, and analysis of three power level transmissions each at three TBO levels. The transmissions featured twin-engine direct-drive configurations of two 500-hp, two 1500-hp, and two 4800-hp power level engines. Each transmission was initially designed for a 1200-hour TBO and was then modified as required to attain a 3000-hour TBO and again for a 6000-hour TBO.

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	ROLE	WT	ROLE	WT	ROLE	WT
Extended Life Transmission						
1200-Hour Time Between Overhaul						
3000-Hour Time Between Overhaul						
6000-Hour Time Between Overhaul						
Twin Turbine Engine Drive						
500-HP Engine						
1500-HP Engine						
4800-HP Engine						

FOREWORD

This report presents the results of a tradeoff study conducted by Bell Helicopter Company (BHC) for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USAAMRDL) during the period 1 July 1970 to 16 April 1971. Study results were attained through the design and analysis of nine transmissions - three power levels and three TBO levels at each power. Design tradeoffs in weight, efficiency, and cost were determined and equated to vehicle performance in the form of cost per operational hour.

The program was performed under USAAMRDL Contract DAAJ02-70-C-0053.

USAAMRDL technical direction was provided by Mr. Wayne A. Hudgins and Mr. James Gomez. Principal investigators were Messrs. R. D. Walker and C. W. Bowen. Acknowledgement for technical contributions is due Messrs. D. Bowers, D. Cleveland, H. Dover, H. Selden, C. N. Warren, and J. Young, all of the BHC Transmission Design Group.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

Figure		Page

LIST OF TABLES

Table		Page

LIST OF SYMBOLS

C, P	Bearing basic dynamic load rating equivalent load
DN	Speed index, bearing bore times rotational speed, MM x RPM
EHD	Elastohydrodynamic
L_a	Number of hours of operation at prescribed power that (100-a)% of a group of similar bearings will complete or exceed before evidence of fatigue develops
L_3, L_5, L_{10}	Number of hours of operation at prescribed power that 97%, 95%, and 90% respectively, of a group of similar bearings will complete or exceed before evidence of fatigue develops
P.D.	Pitch diameter
RVN	Republic of Viet Nam
TBO	Time between overhaul, hours

INTRODUCTION

Helicopter power train components in today's military inventory were designed for Time Between Overhaul (TBO) levels of approximately 1200 hours, but often exhibit Mean Time Between Removals (MTBR) of 40% to 90% of this TBO objective. The improved operational capabilities of these helicopters over prior models in use were in part implemented through the use of low-weight components. In achieving reduced weights, it would appear that the designers have chosen high operating stress limits and have selected low-density materials which are impairing transmission durability and reliability to the extent of increasing the maintenance burden.

The purpose of this report is to present the results of a tradeoff study to evaluate the operational cost of increasing helicopter transmission system TBO levels into the 3000- to 6000-hour range. This evaluation was accomplished through determination of the overhaul life effect on cost, availability, and performance for three distinct sizes of helicopter power transmissions featuring twin-engine inputs of total installed horsepower values of 1000, 3000, and 9600.

The basic parameters influencing design for increased TBO levels were identified for these systems from BHC and Army experience with analysis of failure modes of existing transmissions. Conventional design stress-life relationships and those demonstrated by field operation results were correlated, and suitable design techniques, material specifications, and stress allowables were determined for use in this study. Each of these three horsepower level systems was then designed with objective TBO levels of 1200, 3000, and 6000 hours. For purposes of comparative cost analysis, the assumption was made that these systems are operated in the RVN theater. This assumption, in turn, established the turnaround shipping costs, flight mission spectra, and on-site fuel costs used in calculating the transmission overhaul dollar cost per flight hour - the basis of the cost-effectiveness determination.

PARAMETRIC DISCUSSION

DESIGN PARAMETERS

The proposed transmission configurations for the extended TMO study were predicated on high reliability requirements which dictate the use of a twin-engine installation. Each transmission is driven directly from twin engines of the required horsepower for each size level. The configurations were further restricted to helicopters with one main rotor and an antitorque (tail) rotor. The scope of the study included sufficient horsepower range to encompass light, medium and heavy helicopters: twin engine input at 500 hp per engine, 1500 hp per engine, and 4800 hp per engine The speed range was also chosen to broadly encompass present turbine shaft output speeds and anticipated advanced technology engine (ATE) speeds: 6000 rpm to 24,000 rpm. The following parameters were used in the three horsepower level designs:

Parameter	1st	2nd	3rd
Engines	2 @ 500 hp @ 6000 rpm	2 @ 1500 hp @ 24,000 rpm	2 @ 4800 hp @ 9000 rpm
Trans. Ratio	17.8:1	96:1	60:1
Trans. Design HP	667	2000	6400
Main Rotor RPM	340	250	150
Tail Rotor HP	106	275	880
Tail Rotor RPM	1600	1100	650
Gross Weight	5400	16,000	50,000
Antitorque Trans. Ratio	13:49	19:71	23:63

The tail rotor horsepower listed above was derived from Figure 1, which presumes a straight-line relationship between main rotor horsepower and required tail rotor horsepower. Two points, OH-58A and AH-1G, were used to generate the line. No analysis was made of the relative rotor solidity, main rotor power required, or antitorque rotor power required as influenced by size and power range. This idealization in no way detracts from the results of the study; the tail rotor drive system can be sized for any power level to meet actual requirements.

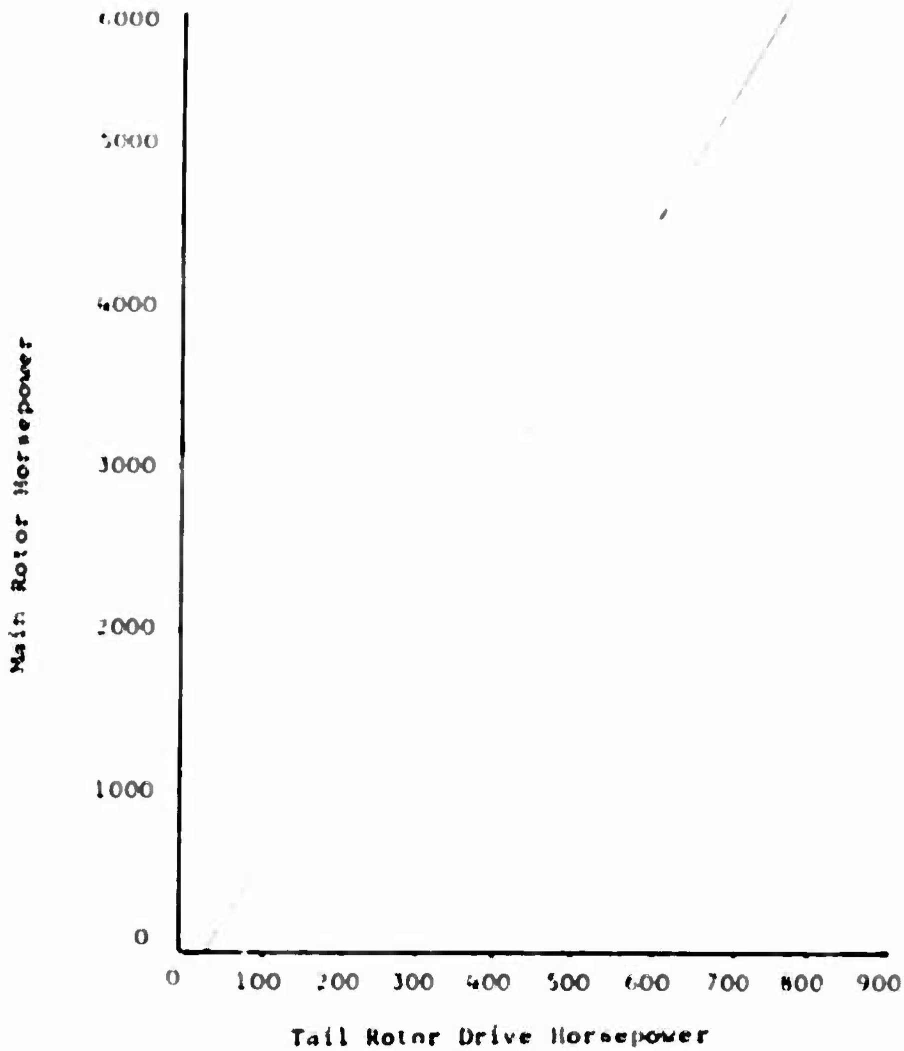

Figure 1. Tail Rotor Drive Horsepower Requirements Vs Main Rotor Power.

Design power conditions for the extended-life transmissions are further derived from recorded flight load spectra obtained on AH-1G, UH-1C, and UH-1H helicopters in the RVN theater. The data acquisition and reduction systems are described in a BHC report (Reference 1). The results of this program were presented in histograms of flight maneuver loads, engine power, main rotor rpm, altitude, and flight speeds for the three models mentioned. In order of severity, it was definitely shown that the AH-1G ranked highest in power requirements, g-loads, and flight speed. Cubic mean power requirements were derived from the AH-1G power histogram and used as a base line for establishing the extended TBO transmission design power. The AH-1G was specifically chosen since it produced conservative (more stringent) design requirements on the extended TBO transmission. A comparison of the operational power data of the AH-1G, UH-1C, and UH-1H is tabulated below:

Model	Average Power (HP)	Normal Engine Power (HP)	Power Available (HP)	Military G.W. (Lb)
AH-1G	795	1250	1400	9500
UH-1C	662	1100	1100	8500
UH-1H	611	1250	1400	9500

As a result of the cubic mean power exhibited by the AH-1G (Figure 2) compared to the normal rated power (827/1250 ≈ 2/3), a design factor of 65% was established for bearing design.

Since the input sections of each transmission may be required to transmit full engine power in the event of single-engine operation, the design power is full engine power available. Thus, in the event of an engine failure, the remaining engine would sustain adequate power to the main rotor mast. The normal operating condition, however, wherein both engines are functional, would not require the full power available from each engine; nor would this be desirable. The helicopter design gross weight must be within the capability of a single engine to sustain flight. This would preclude the use of an engine/gross weight combination that would result in excessive loss of flight capability in the event of a single-engine failure, since 50% power is generally not sufficient to climb at full gross weight. Therefore, the actual main rotor design power is established as two-thirds of twin-engine power, thus providing 75% of this power to the main rotor in the event of single-engine failure. This is adequate to sustain flight within a restricted but safe maneuver envelope.

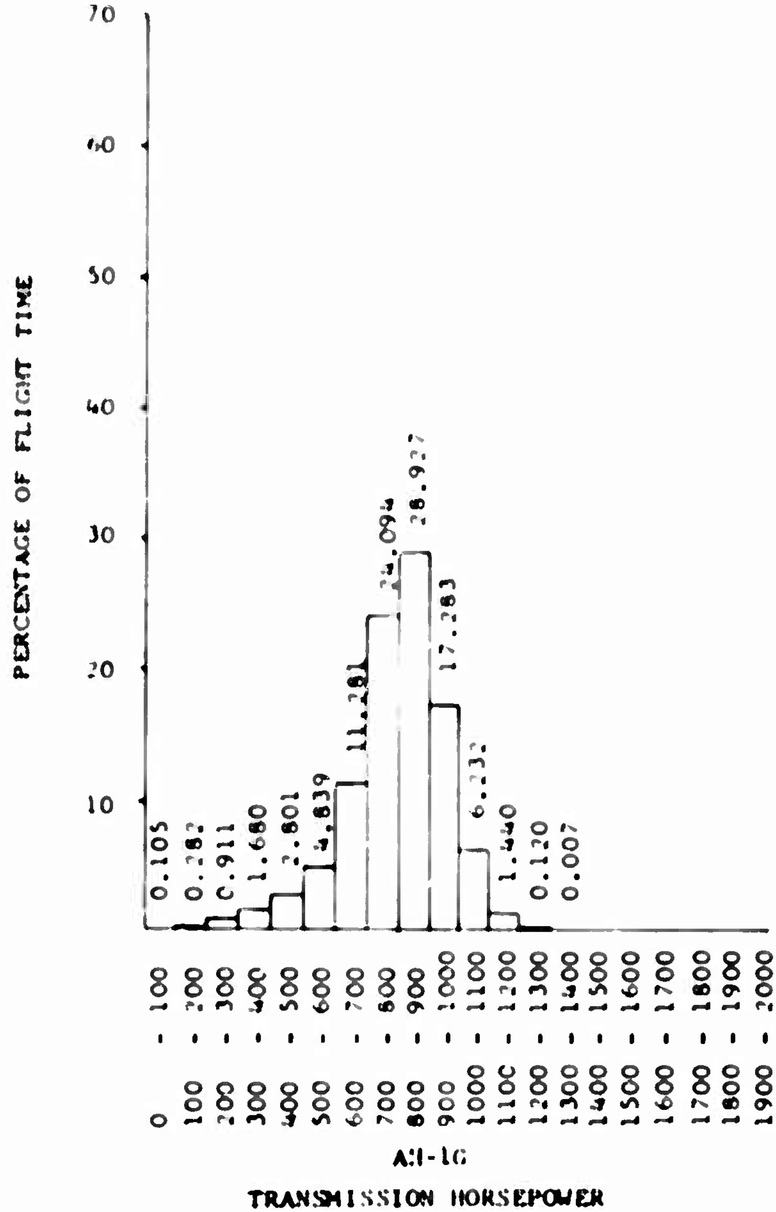

Figure 2. AH-1G Power Histogram.

The following design parameters were established:

1. Gear Design

   A. Input gears are designed with 95% confidence of 99.5% reliability in bending fatigue and 5% probability of pitting for single-engine power.

   B. Planetary gears are designed for the same statistical reliabilities at two-thirds twin-engine rated power.

2. Bearing Design

   Input bearings are designed for a B_{10} life:

   A. 2700 hours at 65% takeoff (T.O.) power for a 1200-hour TBO transmission.

   B. 5000 hours at 65% T.O. power for a 3000-hour TBO transmission.

   C. 7500 hours at 65% T.O. power for a 6000-hour TBO transmission.

   All other bearings must have similar lives, but at 65% x two-thirds twin-engine power.

   A 12% overpower requirement is imposed for design of gears and bearings in the 3000- and 6000-hour gearboxes in excess of power required on the 1200-hour TBO transmission. This additional power requirement is derived from the power spectrum exhibited by the AH-1G helicopter (Reference 1) in the RVN theater, wherein operation at powers above T.O. rating was recorded. Rated engine power for the AH-1G is 1400 hp maximum and 1250 normal rated power with a transmission rating of 1100 hp. The AH-1G actually attained some flight time at power up to 1400 hp.

   Since the power available for twin-engine design is above normal T.O. ratings, as with the AH-1G, it is anticipated that a portion of that power will occasionally be used. The proportionality factor, 1.12, is derived from the normal rated engine power and available power of the AH-1G:

$$\frac{\text{Power Available}}{\text{Normal Rated Power}} = \frac{1400}{1250} = 112\%$$

3. Utilization rate of 50 flight hours per month was chosen.

4. The design of bearings to withstand 3000 and 6000 hours of operation at a preset power level must entail a statistical rate of failure, which results in "premature" removal due to fatigue failure in less than the TBO period. In order to reduce the number of predicted failures, the life requirement, and hence size and/or complexity of the bearing (and gear), is increased above the TBO. Hence, a reasonable statistical failure rate with adequate confidence level must be chosen that will not overly burden the transmission with weight or cost. For this reason, suitable failure rates were chosen as 3% of the 1200-hour TBO transmission, 5% for the 3000-hour TBO transmission, and 8% for the 6000-hour TBO transmission. These choices of statistical percentage of failures result in required design B_{10} lives of 2720 hours for the 1200-hour TBO transmission, 5000 hours for the 3000-hour TBO transmission, and 7500 hours for the 6000-hour TBO transmission. The target design lives thus established afford reasonable statistical reliability while preventing excessively difficult design specifications.

   Under identical power conditions, the probability of a bearing's attaining a low predicted life is much less than the probability of attaining a high predicted life. This is shown subsequently where the generic failure rate drops for increasing TBO, but the basic B_{10} life requirements are higher for extended TBO.

GEAR DESIGN

In general, there are four fundamental failure modes for gear teeth. These are considered to be mechanical wear, breakage (impact or bending fatigue), pitting or spalling, and scoring or scuffing.

The first, simple wear, is generally abrasive in nature and, under conditions of proper protective environment, exists only in relatively soft materials and very low pitch-line velocities. For helicopter transmission system design, the use of hardened (Rc 58 min) gears and pitch-line velocities in excess of 1000 ft/min completely eliminates failure from simple mechanical wear.

Tooth breakage, the second mode of failure, is most generally related to the basic material endurance strength; impact

sufficient to produce brittle failure is not generally found in relatively high-compliance mechanical drives such as found in helicopter systems. In practice, today's limiting design loads tend to be fixed by pitting and scoring phenomena. At such load levels, helicopter transmission spur and helical spur gears can be designed and manufactured with such accuracy and control that bending failure rarely, if ever, occurs. As an example of this fact, over 12,000 UH-1 transmission systems have been built, containing well over 200,000 spur gears, and in all units returned for overhaul, not one instance of tooth breakage has been observed (Reference 2). However, this is not entirely the case with regard to spiral bevel gears. Spiral bevel gears, as manufactured on current machine tools, have inherently inferior root fillet geometry, resulting in higher stress concentration and greater tolerance variation in root fillet radii when compared with conventional spur and helical gears.

For these reasons, coupled with the observation that spiral bevel gears will generally exhibit greater pitting life than spur gears under equal load intensity, design loads are more often than not limited by tooth breakage. The AGMA spur bending stress calculation method and the Gleason Works hypoid and bevel stress method (both computerized at BHC) are adequate design tools to obtain wholly reliable service operation.

Pitting failure phenomena cannot be discussed apart from the scoring and lubrication distress failure modes. In classical gear design practices, the pitting life of gears is related to the Hertzian contact stress by the inverse ninth-power law much as in antifriction bearing life theory. In this approach, the basic material endurance or capacity is proportional to its macroscopic hardness and further related to specific chemistry, nonmetallic inclusion size and frequency, grain orientation, and residual stress field (Reference 3).

In this sense, the pitting defined is essentially pitch-line or rolling-contact fatigue. More often than not, however, other pitting or spalling modes give more trouble in helicopter transmissions. Scoring, discussed in depth later, can lead directly to pitting if the severity is of sufficient magnitude. The scored areas are surrounded with untempered, rehardened (mass quenched) martensite, while the primary scored area may be in a relatively soft tempered or annealed state. Repeated stressings will lead to progressive crack propagation from the rehardened interface until severe pitting occurs. The second alternate mode is generally termed case crushing. The gear face develops severe longitudinal and transverse cracks, which yield to formation of large pits or spalls. This is simply attributed to insufficient case depth to support the subsurface shear stress envelope beneath the Hertzian contact

band. The final pitting mode is usually the most frequent in spur gears operating in synthetic lubricants such as MIL-L-7808 oil at relatively modest pitch-line velocities. It may be treated as an interrelation between elastohydrodynamic lubricant film thickness and tooth surface roughness. The origin of such pits is on the surface at severe asperity contact locations, generally on the driver dedendum at the location of the first point of single tooth contact. The exfoliation progresses in a fan-like shape, broadening and deepening in the direction of sliding (often undermining large sections with subsurface cracks) until large pits or spalls are evidenced. This reduction in normal pitting life due to thin-film lubrication effects has been treated extensively in Reference 3. This type of pitting accounts for 90% of the primary failures observed in closed-circuit overhaul of UH-1 transmission systems (Reference 2).

Successful operation of a properly designed set of gears is finally dependent upon the uniform axial and controlled profile distribution of normal tooth loads and the interdependent conditions of lubrication. Assuming that proper alignment is achieved at the design power deflection condition, the first point of contact occurs at the tip (O.D.) of the driven gear, and contact progresses down the profile until the tooth goes out of action at the driven gear flank. The driven gear tooth assumes a certain amount of the transmitted load immediately upon contact. This load can range from near zero to several times the single tooth transmitted load. Since the teeth are cantilevered beams acting under an elastic loading condition, there is a calculable amount of deflection present. The pair of teeth just preceding the set that is about to contact at the driven gear tip will be deflected such that the net effect will be felt as an index error at the first point of contact. If no attempt is made to relieve this index error, a direct overload will occur. By proper analysis and design, this "mis-indexing" can be eliminated. This generally is accomplished by modifying, or relieving, the tooth profile at the driven gear tip an amount that will be equal to, or slightly less than, the deflection of the pair of preceding teeth. A similar condition of deflection and modification must also exist at the last point of contact, to prevent an overload condition as the teeth go out of action.

There are several types of distress that can occur as an ultimate result of improper tip and flank relief in a heavily loaded set of gears. Generally, the initial distress is scoring which may rapidly progress to destruction of the critical profile shape, leading ultimately to premature pitting, if the load, lubrication, and speed conditions are sufficiently severe. Scoring may be evidenced as bright-polished

radial grooves at the tips and flanks of the teeth, caused by direct metal-to-metal contact in conjunction with the swiping action present at the gear tooth extremities.

Although the exact physics of definition remain the source of much debate, scoring may certainly be attributed to a progressively increasing contact temperature generated by high-relative-sliding, high-unit-load, gear tooth geometry, and constant metallic contact of sufficient energy density to reach the surface liquification temperature of approximately 2900°F. The ability to transmit high torque loads is contingent upon maintenance of a film of lubricant within the contact area that is of sufficient depth to prevent progressive harsh asperity contact between the conjugate surfaces and limit the surface energy density to less than critical values. Providing there are no asperity contacts, or infrequent contacts, the lubricant temperature in the contact area will stabilize and no distress will occur. However, if the speed, lubricant type, and transmitted load are combined so that incipient scoring occurs before temperature stability is attained, then scoring distress is imminent. The design of a successfully operable gear set would then dictate that an adequate lubricant film relative to the surface roughness values of the operating teeth be maintained under all conditions of operation within the design power envelope.

Definition of adequate film thickness must, of course, consider the fact that the total system of lubricant - gear metal reaction in the so-called EP additives can grossly influence apparent critical film thickness ratios. Lubricant film thickness is interdependent with coefficient of friction; in the thick film lubrication region, an increase in temperature decreases coefficient of friction. Thus, the efficiency increases with lubricant temperature. However, an increase in temperature is accomplished by a decrease in lubricant viscosity and film thickness. If progressive asperity contacts occur, the coefficient of friction will increase and the temperature will not stabilize. Scoring will then result (Reference 4). Whether or not this metallic contact is an abrupt result of collapse of the lubricant film at some intrinsic "critical temperature" of the lubricant is an unsettled question in today's gearing technology.

The analytical tools at hand, while not completely general in nature, are quite adequate for engineering design work when based upon extensive experience. For operation in mildly reactive lubricants, such as MIL-L-7808 with the case carburized and nitrided gear steel used in this design, such experience is available. The stress levels to which such gear teeth can be loaded and still successfully operate for

requisite time intervals have been determined by extensive testing of the UH-1 transmission. Based on the test results and with an intimate knowledge of the operating loads and environment, a theoretical analysis has been devised that will satisfactorily predict the instantaneous stresses and loads of a given gear set. With the theoretical analysis as a design tool and with knowledge gained through experience as a guide, a highly reliable transmission system can be designed and manufactured, with a minimum development cycle. The extended-life transmissions have been designed to these known operating limits and represent a reasonable approach to the existent state of the art.

SPIRAL BEVEL GEAR DESIGN

The input spiral bevel gears in each transmission were designed with the aid of a Gleason Company computer program. Three main design features were carefully considered in the bevel gear analysis: safe gear operating stresses consistent with the high-speed application, shaft mounting configuration, and imposed bearing loads.

Safe operating stresses are essential to successful long-life operation of the bevel teeth. It is recognized that the high pitch line velocity ($>$ 10,000 feet per minute) of the twin 1500 and twin 4800 transmission input pinions aggravates the operating stresses due to any dynamic tooth loading condition that exists in the teeth. In order to partially offset the dynamic loading condition and to prevent overstressing, the bevel gear teeth were designed with a high mismatch contact ratio. This, coupled with relatively low calculated bending stresses, assures safe operation (Reference 5).

The mounting system of both the pinion and the gear member materially affects the operating stress conditions in the teeth. All bevel gear members in the main power paths of each transmission are straddle-mounted. This arrangement yields less deflection under load than an overhung gear mounting and provides more accurate conjugate tooth contact, thus minimizing pattern shift and reducing the applied design mounting factor. Internal clearances held in the roller bearings are relatively high due to fail-safe operation requirements, but the contact patterns are well developed through a deflection test program to accommodate known clearance conditions. The inherent rigidity of angular contact duplex and triplex ball bearings mounted as shown also contribute to the accuracy and smooth operation of the bevel gears.

The third item given close design consideration in the bevel gear analysis was the component loads. Unnecessarily high

radial or axial loads can result from improper selection of the helix angle and pressure angle for the bevel teeth. For a given set of tooth numbers, the tangential tooth load varies only with pitch, so a proper choice of helix and pressure angles for the bevel teeth can minimize the reaction loads on the support bearings. This must, of course, be accomplished within the contact ratio limits discussed earlier. Various bevel gear configurations were investigated, and the data for the final selected sets are shown in Appendix I. These sets represent optimum designs, considering the above factors.

Gear Materials

The material for all the external gears is vacuum arc remelted AMS 6265 steel. The gear teeth are selectively carburized to an effective case depth of .030 to .038 inch. The stock removal after carburizing is .001 to .006 inch; hence, the finished effective case depth is .024 to .037 inch. Case hardness is Rc 60-63, and core hardness is Rc 33-41 on the finished gears. The carburizing process (per Bell Process Specification FW-4420, Class A) includes carburizing, subcritical annealing, quenching, deep freezing, and tempering.

The material for the internal ring gears is vacuum arc remelted AMS 6475 steel. The gear teeth are selectively nitrided to an effective case depth of .018 to .024 inch. The stock removal allowed after nitriding is .001 to .005 inch; hence, the finished effective case depth is .013 to .023 inch. Case hardness is R_{15} 90.0 minimum (on ground surfaces), and core hardness is Rc 38-44 on finished parts.

The large size of the gear components used in the twin 4800 probably precludes the attainment of full core properties in the AMS 6265 material (governed by section thickness) and full strength properties of the AMS 6475 (governed by low ratio of case thickness to section thickness). If size effects should produce degradation in mechanical properties sufficient to endanger the extended life design then other materials would be considered, such as SAE 8620 or H-11. However, in the interest of economy the first choice of materials would be as noted above since considerable knowledge of processing exists and material cost is relatively low.

Bearing Design

The B_{10} lives shown subsequently were obtained with the aid of a high-speed digital computer program. Bearings operating at high rotational speeds have appreciable loads generated internally from centrifugal forces and gyroscopic moments, which are calculated by the computer program and included in the statistical treatment of the load-life relationship (Reference

c). Since the bearings supporting the input bevel pinion and gear shafts in the twin 1500 and twin 4800 rotate at very high rpm (24,000 input and 9,000 input, respectively), the internal bearing geometry and associated effects of centrifugal and gyroscopic forces were given close attention. The ball spin/roll axes are greatly influenced by speed-produced internal forces, and rapid catastrophic failure can occur if the controlling race changes at such high speed. Frictional heat causing rapid wear to occur during control transition leads to accelerated failure. These deleterious conditions were precluded by choosing proper preload, race curvatures, and contact angles.

A concurrent bevel gear analysis was conducted to obtain a spiral bevel gear configuration that produced the least bearing loads while maintaining proper contact ratios and allowable stresses. Several arrangements were investigated to attain an optimum bearing bevel gear balance.

In deference to the high-speed application, it is essential that no distress appear in the bearings since the rate of failure progression is too high to ensure detection before catastrophic failure. The normal sequence of events, as experienced with the UH-1 transmission bearings, is to detect a failure, either audibly or by magnetic chip sensor, to isolate the location, and to take corrective action by replacing a quill or the entire transmission assembly. However, with the high-speed twin 1500 and twin 4800 inputs, the probability of having sufficient time for detection and correction before catastrophic failure occurs is greatly reduced. Consequently, the need for high reliability is paramount, and the CEVM M-50 bearing material, as used in the 6000-hour TBO transmission, contributes significantly to this requirement (Reference 8).

The inner rings of the ball bearings are press fitted sufficiently to preclude occurrence of detrimental ring creep and fretting corrosion. The proper fit values can be calculated with excellent confidence by an experimentally derived method developed by BHC. When the elastic rings of duplex and triplex bearings are interference fitted, the inner raceways grow diametrally and induce additional axial preloading when the rings are clamped flush. These bearings must be fabricated with the proper endshake (inner race face intrusion) to secure the desired mounted preload at the high interference fits. The shafting inside diameters are similarly fixed by the strain requirements in the inner rings (at the determined fits) to produce the required interfacial pressures to prevent creep. The resulting mounted preload is highly critical of tolerances. Therefore, the ABEC classes and shaft journal tolerances chosen must be compatible with system design requirements.

For high-speed bearings, the effects of lubrication are such that a marked increase in life can be attained when compared to similar conditions at low speed, primarily due to lubrication influence (Reference 7). Under high-speed conditions, the lubrication regime becomes fully hydrodynamic and the resulting conjunction compressive stress and subsurface stress depart markedly from the classical Hertzian stress. Hertzian stress conditions are probably not attained even at low speed with boundary lubrication, wherein traction stresses become pronounced such that surface shear forces contribute to the production of limiting surface and subsurface stresses. With full hydrodynamic separation in the conjunctions produced by high speed, two physical characteristics prevail: the moving conjunction becomes increasingly sensitive to the raceway sphericity or cylindricity wherein the rolling elements "hit the high spots," and the peak subsurface stresses are diminished and occur nearer the surface. The former condition can induce skidding, then smearing or scoring with attendent heat generation, and then eventual surface fatigue failure or seizure due to loss of geometry.

The latter condition can induce surface-initiated fatigue spalls, possibly from hydraulic action in minor surface imperfections, or subsurface-initiated fatigue spalls of classical origin but very near the surface. In either case, the topological undulations contribute to the failure probability if allowed to become excessive. For these reasons, the contacting element surface conditions relating to osculation and especially waviness require refinement and control to a much higher degree than low-speed bearings. However, surface finish requirements actually become more severe in low-speed bearing applications. It is also recognized that the thin-film, low-speed contacts are far more sensitive to chemical effects in the lubricant-metal conjunction (Reference 8).

AFBMA grade 5 or better bearings are, therefore, necessary for the attainment of target life. Increases in B_{10} of the order of 400% have been recorded when operating seemingly identical bearings at high speed (10,000 rpm) as compared to low-speed operation (1500 rpm) (Reference 7).

The use of a bearing made from vacuum-melted M-50 steel as a direct replacement of a 52100 steel bearing results in a B_{10} increase in the application on the order of 4X or greater. This has been demonstrated at BHC on split inner ring main rotor support bearings run at low speed and high load (Reference 8) in MIL-L-7808 lubricant. The life increase shown by these tests, using similar bearings made from 52100 steel as a base line, was considerably greater than 4X. Similar tests were reported by SKF but at high speed, with the results being

similar to BHC low-speed experience, but life increases only slightly greater than 4X were realized with the M-50 bearings.

The most effective method of preventing inner ring creep in roller bearings is to eliminate the ring and to finish the raceway integrally with the shaft. This practice has been in extensive use on all UH-1 series transmissions, with excellent results. Not only is the tolerance range of internal clearance significantly reduced by this practice, but the fatigue endurance of the carburized vacuum arc remelted gear material far exceeds that of conventional through-hardened bearing materials. Additionally, cost and weight savings are realized by elimination of the inner rings and the associated spacers, nuts, and locking hardware.

The recommended tolerance grade is ABEC 7 for the high-speed ball bearings and ABEC 5 for the roller bearings. Since preload, race control, and press-fitting tolerances of the ball bearings are interdependently critical, it is essential that the tolerance variations be minimized. All journals are to be superfinished to a CLA of 8. Considerable effort must be expended to insure maintenance of the proper dimensions and tolerances, which in turn ensures successful bearing operation.

The maximum life exhibited by a roller bearing is obtained in a roller bearing that is closely controlled dimensionally, has rollers that are crowned for proper load distribution on the rollers, has even roller guiding with close roller and clearance accomplished through integral flanges on either or both races, and, finally, is operated in a lubrication-speed regime that produces relatively thick film separation of rolling conjunctions at the precise load for which the crown was determined (Reference 9).

Relaxation of any of the dimensional or operational characteristics from the optimum results in a reduced life for the bearing. Actual realization of the optimum bearing in addition to continuous operation under the precise design load condition is seldom attained. For this reason, a means of accounting for departure from the optimum must be incorporated in the statistical treatment of life predictions for the roller bearing. This has been done as described in References 6 and 10 (load prediction). Concentrated loading in roller-race conjunctions, especially counterformal, necessarily reduces the exhibited life, compared to the optimum roller bearing. The reduction in life is predicted through the use of a capacity reduction factor, Λ. Tests run on several different configurations of roller and ball bearings resulted in a statistical Λ value of 0.61 for crowned roller bearings and 1.0 for ball bearings. The range of values for roller bearings was 0.45 to

1.0; the lower value of Λ was obtained in straight rollers exhibiting point contact, and the higher value of Λ was attained on bearings fabricated with specially controlled manufacturing processes and dimensional conformity. The Λ factor is influenced by roller guidance, load range, surface finish and waviness, roller crowning, and ring alignment.

Attainment of statistical service life of the bearings used in the nine transmissions of this study relies on the use of high-quality bearings (AFBMA Grade ≥ 5), wherein a minimal value of Λ = .61 is assured. With proper load predictions and operating DN values, a higher value of Λ is justified. For example, the roller bearings in the high-speed input bevel pinions of the twin 1500-hp transmission are analyzed with Λ = .61. This is done since the speed range and loading condition are such that the effects of manufacturing tolerances, thermal distortion, and roller dynamics are magnified to the extent that realization of optimum life is improbable. The roller bearings on the input bevel gear shaft are also in this category, since the speed is still high (Reference 9).

The tail rotor spur takeoff gear of the twin 1500-hp transmission is mounted between two cylindrical roller bearings in such a manner that end loading of the rollers is allowed. During power oscillation in the tail rotor drive system, it is possible for these rollers to absorb a fluctuating end load and thus increase the possibility of adverse skewing effects which are detrimental to the life of the roller bearing. For this reason, the value of .61 for Λ is used in sizing the bearings.

The output spiral bevel gears in the twin 1500-hp tail rotor drive system are mounted in such a way that a higher value of Λ is justified. The rollers are loaded in the proper C/P range, operate in the intermediate speed range, are rigidly housed, and are properly held in position by the duplex ball bearings. For these reasons, a Λ factor of .83 is used in sizing these roller bearings. The value of Λ = .83 coincides with the AFBMA capacity prediction for roller bearings with modified line contact and integral guide flanges, which assure even load distribution.

The value of Λ for ball-race conjunctions is 1.0. This is not to say that ball bearings have a higher load capacity; rather, it is a function of the mathematical predictability of stresses in a point contact conjunction. Conjunction between balls and raceways, both conformal and counterformal, are always point contacts and hence require no reduction from the optimum for observed conjunction overstress. The conjunction area is elliptical, and the stress patterns are well defined analytically (Reference 11).

The characteristic failure rates are an index to the relative reliabilities for each gearbox design at each of the three TBO levels. Inability to size bearings to provide exact required lives in conjunction with the basic design assumptions yields different levels of reliability when considered from a single level of service life or from the three TBO levels. For example, if each TBO level were analyzed with a predicted failure rate based on 1000 hours, the 1200-hour TBO transmission would have by far the highest failure rate. But when compared to 3000 hours and 6000 hours, the failure rates are more nearly equal, thus indicating proper sizing of dynamic elements.

So in addition to the 1000-hour basis for generic failure rate, we calculate failure rates thusly:

$$\lambda_{\substack{1200\\TBO}} = \frac{1200}{(L_{50})}, \qquad \lambda_{\substack{3000\\TBO}} = \frac{3000}{(L_{50})}, \qquad \lambda_{\substack{6000\\TBO}} = \frac{6000}{(L_{50})}$$

The relative reliabilities simply point out the consistency in design criteria. The criteria are actually different regarding calculated L_{10} requirements, but the failure rates as a function of TBO period show a tradeoff characteristic in favor of extended TBO.

DIAGNOSTICS OR FAILURE DETECTION SYSTEMS

Diagnostics cannot prevent premature failure, nor can diagnostics be expected to eliminate secondary failure. However, diagnostics in conjunction with such measures as compartmentalization can reduce the possibility of catastrophic failure due to secondary damage. For example, a mast bearing spalling failure generates metal flakes that, coursing downward through the transmission, can be entrapped in gear meshes or in bearings. Entrapment in gears could cause excessive deformation or tooth fracture and loss of drive. Entrapment in bearings could ultimately cause ring fracture or loss of operating clearance and subsequent overheating with attendant seizure or loss of drive. The mast bearing failure is primary but not catastrophic in its early stages due to a low progression rate. Seizure or loss of drive in this example is due to secondary causes in the higher speed portions of the transmission, where progression rates are significantly higher.

Transmission reliability is increased by the use of a positive failure detection system. If failures were detected early and isolated so that secondary debris damage would be minimal, then overhaul cost would be reduced. However, the detection system, per se, would have no effect on extended TBO. The

entire question of the cost-effective overhaul point - increased cost of overhaul due to extensive secondary damage vs loss of residual useful life is in fact outside the scope of this report.

Input One-Way Clutch

The function of the input one-way, or freewheeling, clutch is to transmit engine torque to the transmission during normal operation, yet to allow free rotation of the transmission and other rotor-drive components in the event of engine stoppage. A full-phasing clutch manufactured by the Spring Division of Borg-Warner Corporation was chosen for each transmission.

Sprag design of these units is normally such that the clutch is centrifugally engaging. That is, centrifugal force acting at the center of gravity of each sprag rotates the sprag into more intimate contact with the inner and outer races. This feature ensures continuous sprag-race contact and allows smooth engine reengagement, at speed, after a period of freewheeling.

Clutch lubrication is provided by a .28-gpm oil flow rate on the 500-hp unit and a .5-gpm value on the 1500- and 4800-hp units into the end of the inner rings. This quantity of oil lubricates both clutch assembly radial bearings in addition to the sprag clutch proper. Adequate wear life for the increased TBO units is ensured through use of inner-race overrunning designs (wherein the centrifugal load component reduces to zero with a stopped engine) and extra-width silver-plated beryllium copper drag springs.

The bearings in each freewheeling unit serve to center the inner and outer clutch raceways, maintaining proper alignment. There is no relative rotation of the bearing rings during normal power-on operation. Adequate false brinelling resistance to the imposed input drive shaft loads is provided by use of maximum-static-capacity, deep-groove, fractured ring-type bearings. During autorotation or one-engine-out operation, the bearing operates normally with inner ring rotation relative to the outer ring. However, since no calculable loads exist, no life is shown for these elements.

DRIVE SYSTEM FLEXIBILITY

Drive system flexibility has been incorporated into each transmission for the purpose of ensuring proper load sharing in the planetary reduction gears. The ring gears in each case are splined loosely to the housing. This allows the ring gear to deflect radially and circumferentially at each planet passage

and allows the separate planet gear meshes to seek an equal transmitted load. The sun gears are unrestrained radially to provide the same load-sharing tendency by seeking equilibrium within the planet meshes. The lower planetary carriers are designed so as to absorb the torsional windup within the planetary stage and thus prevent gear tooth end loading. This higher speed planetary stage is generally most susceptible to scoring failure when end loading is permitted.

In the low-speed upper planetary assemblies where space limitations prevent the flexible design used in the lower planetary assemblies and scoring risk is greatly reduced, the gear teeth themselves are modified in the lead direction in such a manner that no end loading exists under normal operating conditions. The carrier deflections exist in the upper planetary carriers, but this deflection is anticipated and accommodated.

Excessive flexibility is largely eliminated in the drive system, where deleterious effects may result. This is noticeable in the spiral bevel gear mounting and bearing housings. The nature of conjugate action in spiral bevel gear teeth dictates relative rigidity in mounting to preserve the developed contact pattern. Although recent designs employing relatively high helix curvature (small cutter radius) exhibit decreased sensitivity to deflections, shaft deflections and housing deflections allow relative motion of the gear and pinion which generates a concentrated conjunction load on an abbreviated area of the active tooth profiles. The resulting stress magnitude can be considerably higher than nominal calculated stresses and can lead to surface fatigue failures well in advance of predicted gear life. Bearing sizes and internal clearances, as well as shaft section modulii, can also contribute to gear distress through excessive deflections. For these reasons, the shafts, housings, and bearings must be sized and proportioned to effectively eliminate excessive deflections.

One major contributor to distress from deflections is the main rotor mast. This element can impose radial load to the planetary assemblies when mast shear loads are generated in flight. The bending modes are such that relative radial motion of the mast at the planetary carrier level will induce radial reaction loads in the planetary gear teeth. To circumvent these undesirable deflections, the upper carrier is flexibly splined to the mast above the planetary assembly level. Deflections are minimal at this level, and spline flexibility is helpful in further reducing the effects of deflections.

Housings are so designed that concentrated pylon mounting loads are distributed to the transmission supporting structure

without inducing high local deflections which would affect bearings and gear meshes adversely. This requires that transmission cases be "beefy" and hence contribute to the overall weight. Successful gear and bearing dynamics, however, depend upon the practice of restricting flexibility to that extent sufficient to assure adequate load distribution in elements designed to share load.

EROSION

Erosion of Oil Passages

Careful examination of several cored oil passages in UH-1 magnesium sump cases and steel jets after return for overhaul has revealed no indications of erosion. The UH-1 transmission oil pump is a 12-gpm constant-displacement Gerotor that delivers oil at 60 psi at a cored passage velocity of 30 ft/sec. It is apparent from the investigation that the range of flow and pressure involved is not sufficient to cause erosion damage to the oil passages. One unused case was sectioned and surfaces were compared to verify that the condition of the used cases was rot markedly different from the new cases. Therefore, erosion is considered to present no problem in extending TBO, so long as pressure and velocity experiences are not exceeded.

3000-HOUR TBO TRANSMISSION DESIGN

A 3000-hour TBO is attained through bearing changes wherein sizes are increased to provide the higher capacity required for extended fatigue life; increased gear sizes to accept additional load as needed from power available in excess of main rotor design power and additional gear load cycle accumulation during extended operation; material changes necessary to provide adequate corrosion resistance for extended operation; and seal material and configuration changes to prevent deterioration and wearout.

1. Increased bearing sizes are necessary to achieve sufficient fatigue life for reliable 3000-hour TBO. The size increase was predicated on the reduced allowable operating stresses of the gears, which dictated an increase in horsepower transmitted to the main rotor mast and the L_x failure rate acceptable during 3000 hours of service life. In the case of the 1200-hour TBO design, $L_x = L_3 = 1200$ hours, allowing a 3% failure rate for a population life of 1200 hours. However, for a 3000-hour TBO, L_x was chosen as $L_x = L_5 = 3000$ hours, allowing a 5% failure rate for a population life of 3000 hours. This was done in an effort to reduce the considerable size (hence weight) increase in

attaining 3000 hours TBO. That is, an L3 design of 3000 hours results in excessive bearing size and weight and provides too stringent a failure rate to be commensurate with associated gear life.

2. The input bevel gears and/or helical herringbone gears must be designed for infinite fatigue life at full engine output torque. The gear tooth load cycle accumulation rate is sufficient to produce in excess of 10^7 cycles during the 1200-hour TBO period. Therefore, these gears were designed for infinite life for the 1200-hour TBO and retained for all TBO levels.

   The operating surface compressive stress must remain below some established level to preclude surface fatigue. This stress level must provide margin for transient operation at overpower conditions which could induce surface failure during the extended life cycle. The restriction of operating surface compressive stress to this safe established level is predicated on EHD film thickness and the exhibited fact that gears, under thin-film lubrication, are surface-fatigue-life limited rather than flexure limited. The EHD film thickness, in turn, is influenced primarily by lubricant viscosity at operating temperature. The use of synthetic lubricants and their relatively low viscosity (3-5 cp) necessitates designing to relatively low surface contact stress levels in the low-speed drives, which, in turn, results in relatively "heavy" gears.

3. Material properties play a paramount part in the attainment of extended service life. The operating environment can render a mechanism useless unless the components are specifically designed and treated to withstand the elements of the particular environment. The transmission cases are naturally the components that are most subject to corrosion, especially areas that tend to entrap moisture or retain dirt and debris. The case material must have an inherent tendency to resist corrosion over long periods of exposure, or the transmission will be removed "prematurely" and attainment of the projected TBO will not be accomplished. The "corrosion life" of certain UH-1 transmission cases is limited to two overhaul cycles, and this only after rework to remove the evidence of corrosion at the first overhaul. The cases fabricated from magnesium are treated with corrosion preventative compounds and externally coated with organic coverings to provide the best corrosion protection possible with

the existent technology. However, during field service exposure, the galvanic corrosion around joints where dissimilar metals have to be used (steel studs and washers and magnesium case) reaches an advanced state in 1200 hours (Reference 2). The case geometry is not such that actual water traps are present, but the surfaces around joint studs are flat and horizontal so moisture naturally collects and does not run off. The result is eventual corrosion in spite of external organic coatings and sealers.

There are materials suitable for transmission cases however, and this was shown in Reference 2. All transmission cases used on the Vertol CH-47 are made from aluminum alloy. It was quite evident that no corrosion problem existed with the CH-47 transmission cases, to the extent that no cases were replaced for corrosion during the study period in Reference 2.

The ability, therefore, to resist corrosion in field service is attained primarily through the use of aluminum alloy transmission cases and judiciously designed external geometry to prevent moisture retention on case surfaces. For the 1200-hour TBO transmission design, the cases were fabricated from magnesium alloy, but attainment of TBO of 1200 hours is assured. For the 3000-hour TBO transmission, however, the material was changed to aluminum alloy, and the joints between cases were designed with studs extending downward, thus eliminating moisture-retaining flat surfaces on the exposed upper surfaces of the cases (reference 3000 TBO, twin 1500 top case). The trade off in this instance is adequate corrosion resistance to attain extended service life for the difference in weight in changing from magnesium alloy to aluminum alloy.

4. Shaft sealing for extended service life must be addressed from two aspects: seal wear and aging deterioration. The use of conventional lip seals made from elastomeric materials provides adequate life for the "normal" 1200-hour overhaul period. However, for extended life, the problem of simple elastomer life (shelf life) can render the seal ineffective, generally due to hardening. The seal must maintain a minimum pressure on the shaft in order to provide effective sealing. When the seal material becomes hard or stiff due to aging, the ability to maintain adequate pressure is lost. Therefore, with shaft or sealing ring wear

in conjunction with elastomer hardening, the seal may begin to leak.

For extended life, the shaft seal must be fabricated from materials that resist aging deterioration. This requires the use of face seals, e.g., carbon on steel (generally stainless steel). The carbon face seal resists aging deterioration. This leaves only wearout to be alleviated. Sealing surface wear is not prevented, but an adequate wear life can be attained by proper design and manufacture of the seal, shaft, and housing. Proper design is such that positive lubrication for the seal is provided during operation. That is, it is necessary for the seal to weep or leak a small amount in order to seal effectively and still provide extended wear life. The seal rubbing interface must be separated by an oil film to eliminate wear, which leads directly to seal leakage. Therefore, leakage must be tolerated, but the amount can be controlled (and effectively scavenged from the area). The seal rubbing pressure must be controlled and maintained constant throughout the life of the seal. This is done by proper choice and application of seal springs. The springs are designed to maintain an approximate constant force by making the effective spring length sufficient to minimize the loss of force due to spring extension. In addition to oil seal leakage which provides wear resistance, it is necessary to prevent abrasive wear due to entrapment of dust, dirt, and moisture from the external side of the seal. External protection is provided by a labyrinth seal on the shaft outside the face seal. Dust and dirt are effectively excluded during operation, and moisture, either from rain or cleaning hoses, is excluded statically. The moisture that may enter the labyrinth seal escapes through a drain line provided for this purpose (and for the purpose of dumping weepage oil over the side), so the face seal absorbs no water, thus reducing the probability of corrosion.

Attainment of 3000 hours of operation on the labyrinth-face seal design is reliably assured with the above design features. Another possibility for extended life is in the use of pneumatic seals. Air under controlled pressure can be introduced into a labyrinth on a rotating shaft. This provides a pressure head which effectively eliminates oil leakage past the labyrinth. Bleed air can be supplied from the engine to provide the pressure head, or a shaft-mounted centrifugal blower can be provided. The advantage of this seal

is the absence of shaft-to-seal contact, thereby eliminating seal wear. One disadvantage arises from the secondary support system (air supply) required for proper sealing. Another, and more important, disadvantage lies in the fact that no static seal is present to exclude moisture and debris from the gearbox. For this reason alone, face seals were chosen for extended life.

6000-HOUR TBO DESIGN

The design of the 6000-hour TBO transmission is accomplished through modification of the 1200-hour TBO transmissions to receive the 3000-hour TBO gears (which were designed to reduced allowable operating stresses), bearing material changes, and case material changes.

The gearbox cases are made from cast aluminum alloy, as were the 3000-hour TBO cases. However, the housing proportions around bearings were reduced back to the 1200-hour TBO size since the bearings are the same size as the 1200-hour TBO design.

By changing the 1200-hour TBO bearings from 52100 steel to M-50 steel, a capacity increase was realized that was greater than the capacity increase attainable by increasing the 3000-hour TBO bearings to the next larger size. The life increase was sufficient to achieve the minimum required B_{10} life of 7500 hours for the 6000-hour TBO design.

The gears which were designed to operate at the reduced allowable stresses for the 3000-hour TBO were also used in the 6000-hour TBO designs. The operating conditions for the 6000-hour TBO are more nearly equal to the 3000-hour TBO with respect to the transient or momentary overpower condition than when comparing the 3000-hour design to the 1200-hour design. The cycle accumulation at overload conditions would remain relatively insignificant compared to the billion-cycle tooth contact accumulation. Although some allowable stress reduction for contact fatigue and bending fatigue in going from four billion to eight billion cycles (highest frequency mesh in the twin 500-hp design) is generally considered applicable, little confidence exists in its exact definition.

The question of seal design for extended life beyond 3000 hours must be solved by means other than simply hoping that a face seal will last 6000 hours. The method used in ensuring 6000 hours operation would be to design the seals into quills and replace each seal quill at the 3000-hour interval. Provision to facilitate seal quill removal would have to be made, i.e.,

output or input shaft design that would allow seal quill removal without removing the associated gear quill. However, the seal replacement would be accomplished at the field maintenance level with no special tools required.

External flexible lines used in transferring lubricating oil from transmission to oil cooler and back would also be replaced at the 3000-hour interval. No provisions would be made for quick disconnects since the lines would be normally drained in static position, and replacement frequency would be minimal. This requirement would be completely eliminated by an integral oil cooler design which would preclude external oil lines. Oil system sight gages would be retained for the total TBO, but material for these elements would be heat-treated glass conventionally mounted on O-rings and retained by snap rings.

Externally exposed shafting would be made from corrosion-resisting steels and would be cadmium plated as were the 3000-hour TBO designs. Internal corrosion protection would be the same as the 3000-hour TBO design.

DESCRIPTION OF TRANSMISSIONS

TWIN 500-HP TRANSMISSION

The twin 500-hp transmission is a two-stage reduction unit comprised of two input spiral bevel pinions (Figure 3, item 1G) driving a single spiral bevel gear (Figure 3, item 2G), which in turn drives a single planetary stage. An overrunning clutch is positioned at each pinion location to allow single-engine operation or twin-engine operation with different power inputs.

The input spiral bevel gear drives the lower sun gear (Figure 3, item 3G) through a splined connection. The sun gear drives three planet idlers (Figure 3, item 4G) inside a semifixed ring gear (Figure 3, item 5G). The planet carrier drives the main rotor mast through a splined connection.

The oil system consists of a constant-displacement 6-gpm pump driven from the bevel gear member. Oil is drawn through a coarse filter screen from the main case and pumped through a 10-micron filter to the oil cooler. The oil cooler is mounted above the port input shaft coupling, which has a centrifugal blower attached to the coupling drive flange. Cooling air is furnished to the cooler during normal operation by the blower. In the event of port engine failure, the blower becomes inactive; however, the cooling requirement is reduced due to the power decrease (500 hp maximum), so the inoperative blower is not detrimental to the continued single-engine performance. The justification for this assumption is based on past performance of the UH-1A helicopter transmission, which was designed for 700 hp continuous, and no oil cooler was used for the oil system on the main transmission. The normal operating horsepower at 65% would be comparable to the single-engine operation of the twin 500-hp design of this study. The tail-rotor-drive spiral bevel pinion (Figure 3, item 6G) is driven by the "bull" spiral bevel gear. The tail-rotor-drive output shaft also provides a rotor brake assembly mounting. The shaft rotates at 6000 rpm (same as engine input) and transmits 108 hp to the antitorque tail rotor at 1600 rpm through a 3.75 ratio 90° spiral bevel gear set in the tail rotor gearbox (Figure 3, Tables I and II; Figure 4, Tables III and IV; Figure 5, Table V).

Cases are made from cast magnesium except for the top case, which is made from forged aluminum alloy, 4032. The top case is made from aluminum in order to withstand the fretting wear associated with the external spline on the ring gear.

Conventional elastomeric lip seals are used in sealing input and output shafts. Shaft sealing surfaces are case hardened and cadmium plated.

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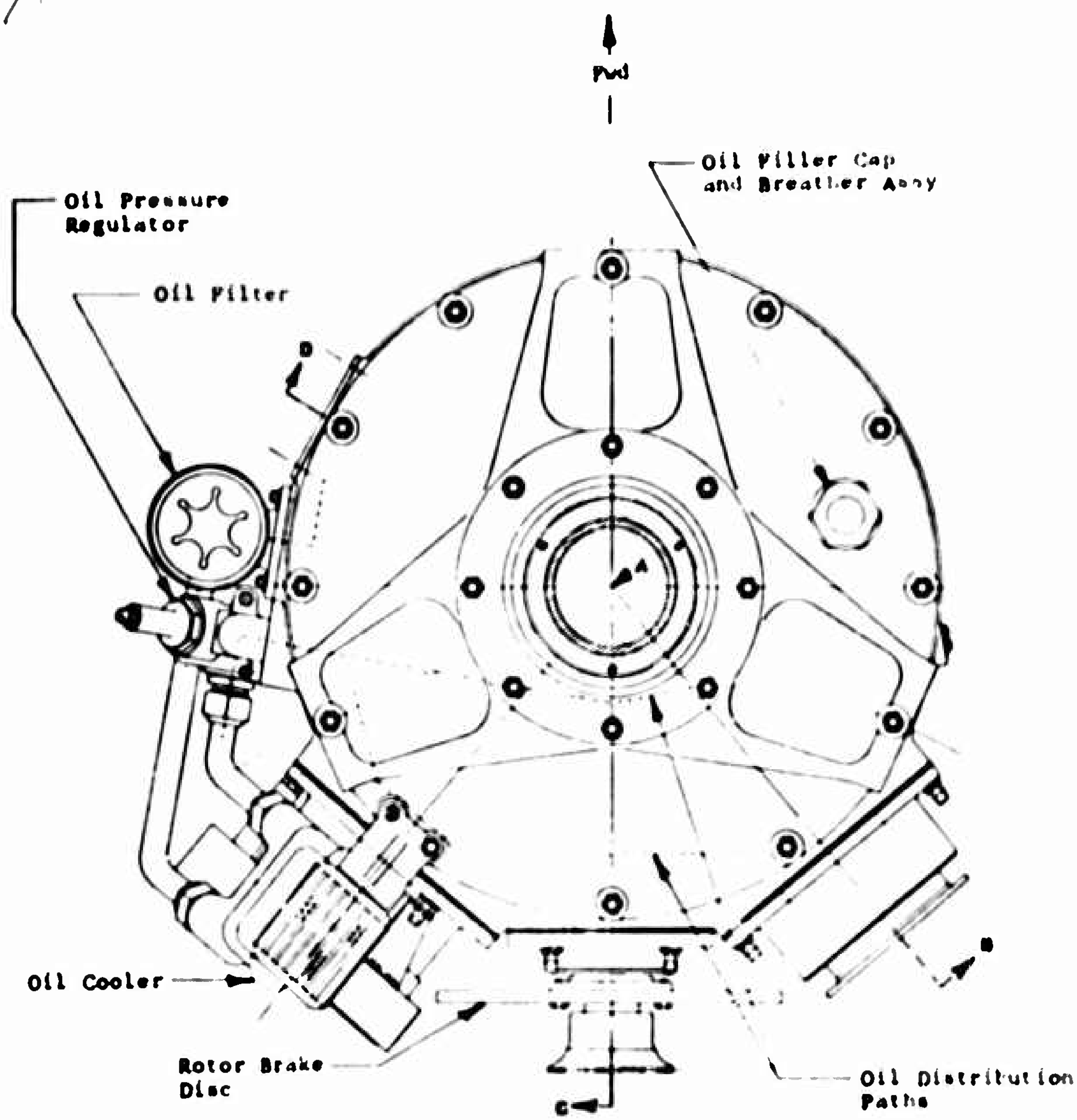

Figure 3. Twin 500-HP Transmission, 1200-Hour TBO.

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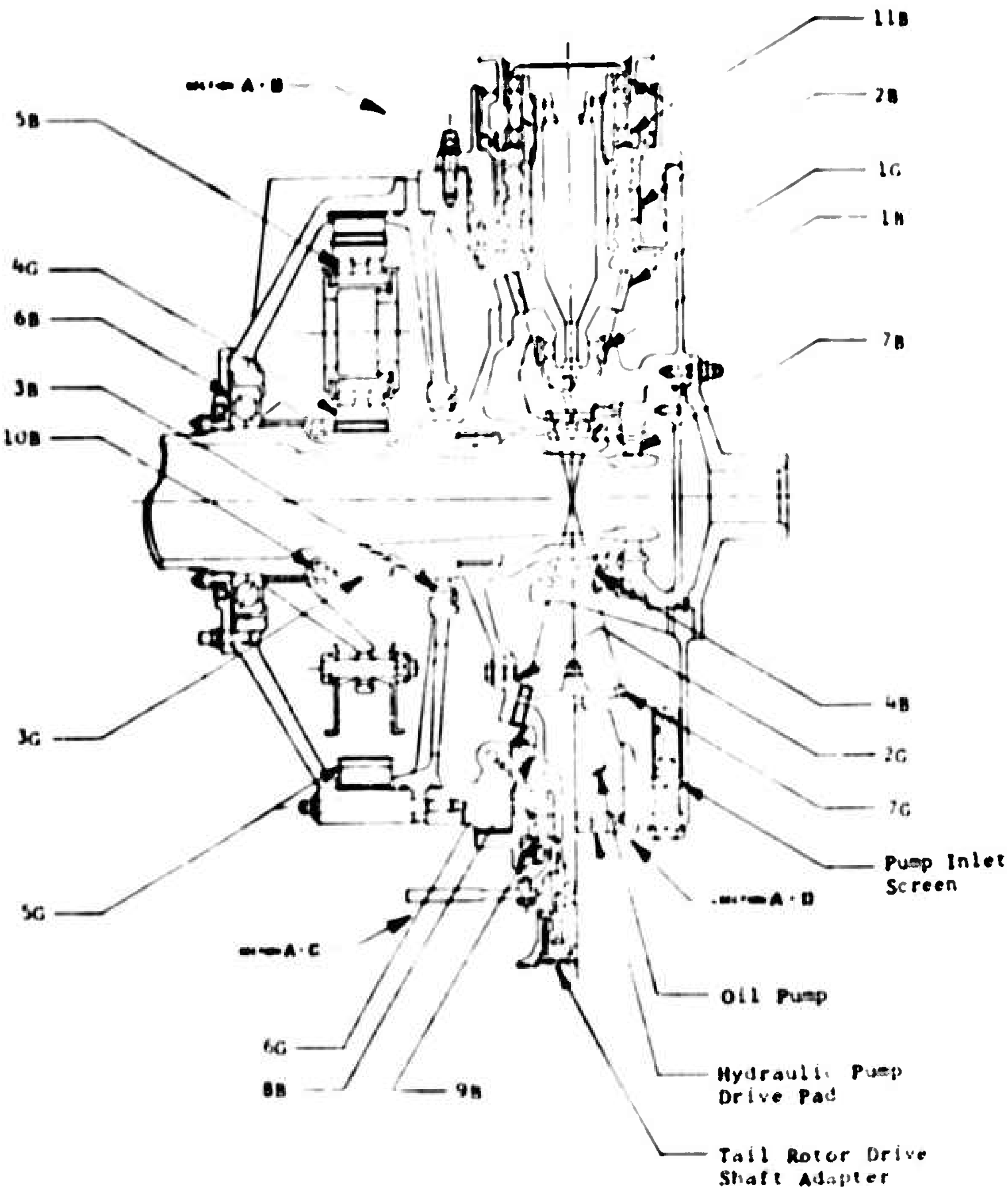

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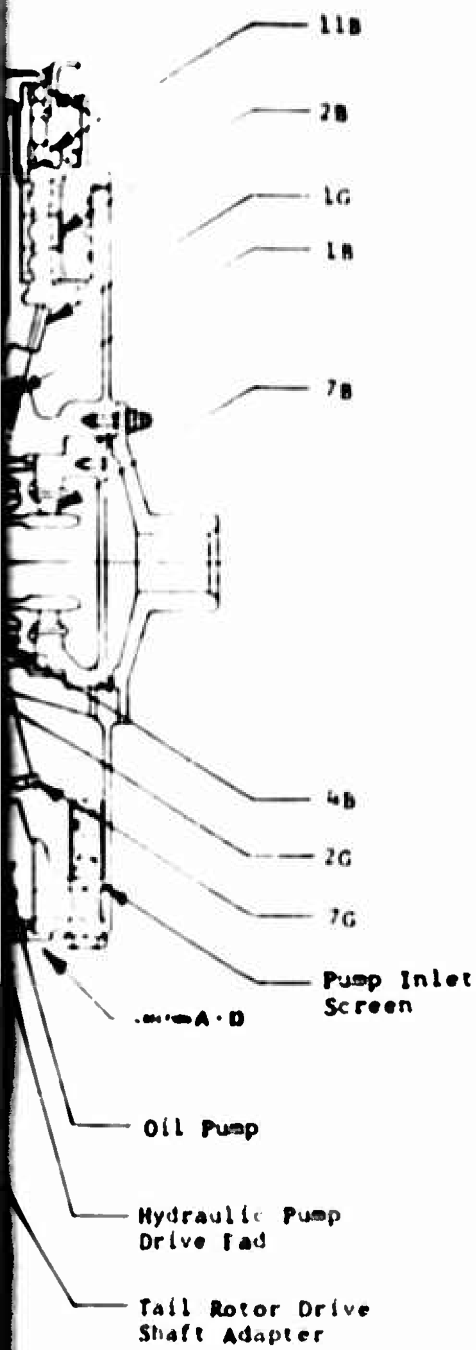

GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Sun Gear
4G	Planet Pinion
5G	Ring Gear
6G	Tail Rotor Drive Pinion
7G	Accessory Drive Pinion

BEARINGS

Item No.	Name
1B	Input Pinion Roller Size 304
2B	Input Pinion Ball Size 7212
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10
4B	Input Gear Ball Size 6918 With 13/32 Balls
5B	Planet Roller Size 11 x 11 x 3.86 P.D. 2 Rows: 16 Rollers/Row
6B	Mast Ball 204-040-136
7B	Mast Roller Size 1011
8B	Tail Rotor Pinion Roller Size 1911
9B	Tail Rotor Pinion Ball 204-040-143
10B	Planetary Support Ball Size 95 x 125 x 13
11B	Freewheeling Ball Size 70 x 100 x 12

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TABLE 1. SUMMARY OF BEARING LOADS AND LIVES

Item	Bearing	Loading * Axial	Radial (lb)	Moment (in.-lb)
1B	Input Pinion Roller Size 304	-	634	-
2B	Input Pinion Ball Size 7212	1,287	904	1,087
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10	-	1,636	-
4B	Input Gear Ball Size 6918 with 13/32 Balls	1,175	703	1,032
5B	Planet Roller 2 Rows: 16 Rollers/Row Size 11 x 11 x 3.86 PD	-	2,100	-
6B	Mast Ball 204-040-136	5,370	4,202	6,734
7B	Mast Roller Size 1011	-	3,509	-
8B	Tail Rotor Pinion Roller Size 1911	-	918	-
9B	Tail Rotor Pinion Ball 204-040-143	623	321	214
10B	Planetary Support Ball Size 95 x 125 x 13	-	-	-
11B	Freewheeling Ball Size 70 x 100 x 12	-	-	-

* 65% Load for Items 1B through 5B
75% Load for Items 8B and 9B
100% Load for Items 6B and 7B
No calculations made for Items 10B or 11B

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EARING LOADS AND LIVES - 1200-HOUR TBO, TWIN 500 HP

ding * dial lb)	Moment (in -lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
634	-	4,445	6,000	11	120,000
904	1,087	2,824	6,000	26	360,000
,636	-	17,026	1,575	11	173,000
703	1,032	4,180	1,575	11	142,000
,100	-	4,568	968	6	73,000
,202	6,734	2,902	337	5	37,000
,509	-	3,754	337	2	19,000
918	-	5,376	6,000	17	330,000
321	214	6,048	6,000	18	210,000
-	-	-	-	-	-
-	-	-	-	-	-

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TABLE II. SUMMARY OF FUNCT
TWIN 500 HP, 120

Item No.	Item	(HP)	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Li Velocit (ft/min
1G	Input Spiral Bevel Pinion	500	3,287	6.0	21	1.2	6,000	5,498
2G	Input Spiral Bevel Gear	500	3,287	6.0	80	1.2	1,575	5,498
3G	Sun Gear	667	4,100	8.29762	36	1.6	1,575	-
4G	Planet Pinion							
	With Sun	667	4,100	8.29762	46	1.6	968	1,404
	With Ring	667	4,100	8.70139	46	1.6	968	1,404
5G	Ring Gear	667	4,100	8.70139	132	1.55	-	-
6G	Tail Rotor Drive Pinion	108	682	6.034	21	.60	6,000	5,467
7G	Accessory Drive Pinion	-	-	6.774	21	.28	6,000	4,870

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B

MARY OF FUNCTIONAL GEAR DATA,
IN 500 HP, 1200-HOUR TBO

RPM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss	Number of Meshes
000	5,498	284.0	37,540	211,758	7.2	2.174	2
575	5,498	284.0	37,613	211,758	7.2	2.174	2
575	-	100.9	44,048	169,712	16.41	.512	3
968	1,404	100.9	50,974	169,712	16.41	.512	-
968	1,404	26.1	32,385	135,470	17.91	.413	-
-	-	26.1	46,235	135,470	17.91	.413	3
000	5,467	212.0	21,761	144,629	8.7	.453	1
000	4,870	-	-	-	-	-	1

A

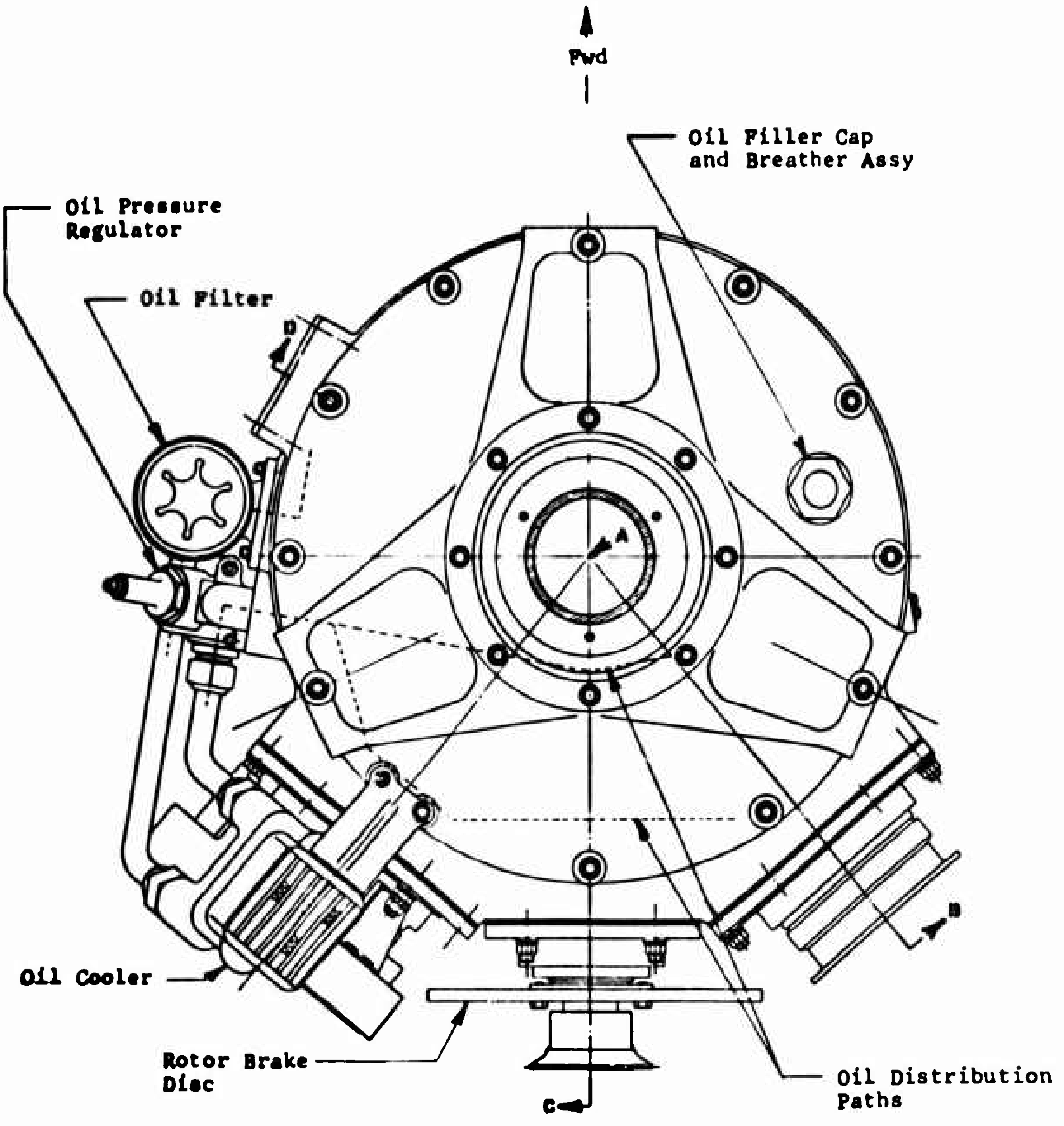

Figure 4. Twin 500-HP Transmission, 3000-Hour TBO.

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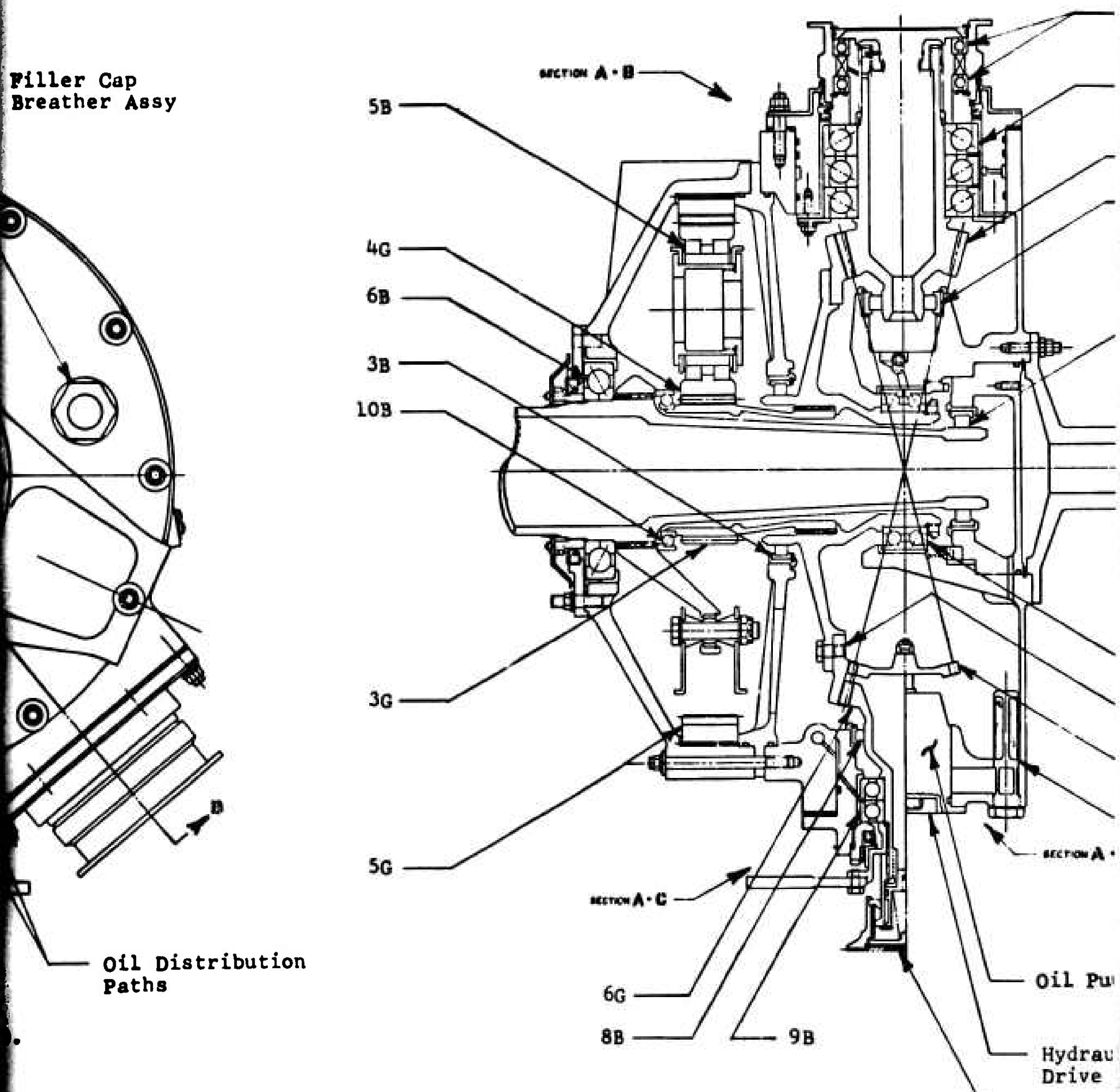
B

Filler Cap
Breather Assy



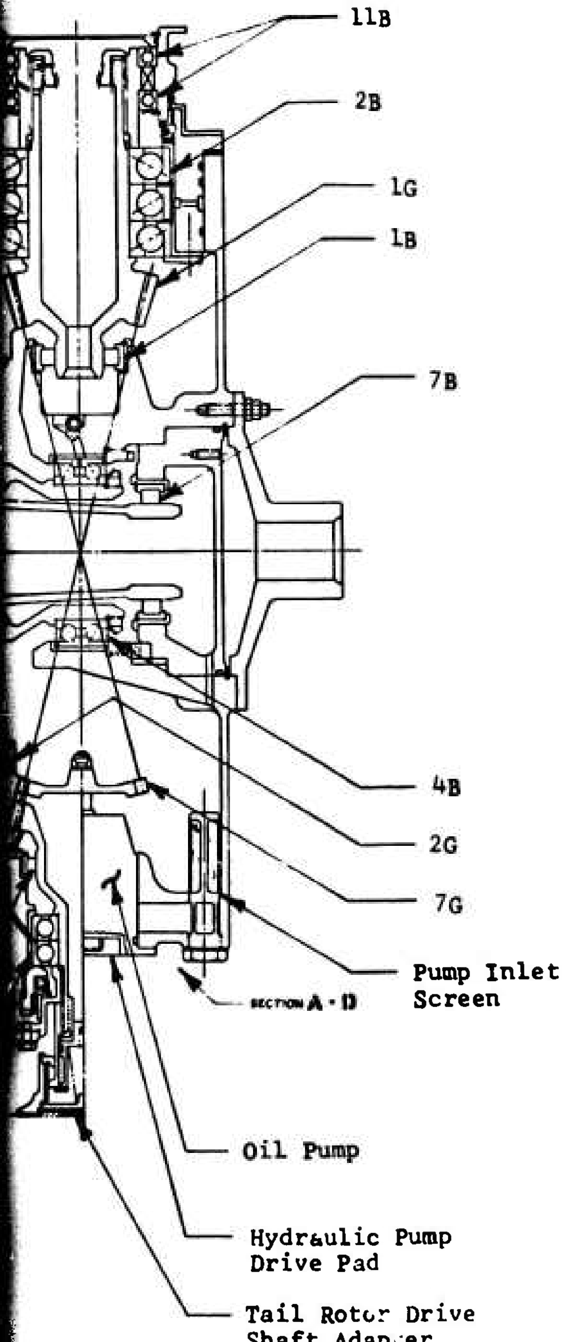
Oil Distribution
Paths

Oil Pu

Hydrau
Drive

Tail R
Shaft

C



GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Sun Gear
4G	Planet Pinion
5G	Ring Gear
6G	Tail Rotor Drive Pinion
7G	Accessory Drive Pinion

BEARINGS

Item No.	Name
1B	Input Pinion Roller Size 305
2B	Input Pinion Ball Size 7214
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10
4B	Input Gear Ball Size 6918 With 7/16 Balls
5B	Planet Roller Size 11 x 13 x 3.86 P.D. 2 Rows: 17 Rollers/Row
6B	Mast Ball 204-040-136
7B	Mast Roller Size 1011
8B	Tail Rotor Pinion Roller Size 1912
9B	Tail Rotor Pinion Ball 204-040-143
10B	Planetary Support Ball Size 95 x 125 x 13
11B	Freewheeling Ball Size 70 x 100 x 12

A B

TABLE III. SUMMARY OF BEARING LOADS AND LIVES

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Be
1B	Input Pinion Roller Size 305	-	834	-	
2B	Input Pinion Ball Size 7214	1,279	732	999	
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10	-	1,818	-	
4B	Input Gear Ball Size 6918 with 7/16 Balls	1,135	665	819	
5B	Planet Roller 2 Rows: 17 Rollers/Row Size 11 x 13 x 3.86 PD	-	2,353	-	
6B	Mast Ball 204-040-136	5,370	4,202	6,734	
7B	Mast Roller Size 1011	-	3,509	-	
8B	Tail Rotor Pinion Roller Size 1912	-	977	-	
9B	Tail Rotor Pinion Ball 204-040-143	632	365	240	
10B	Planetary Support Ball Size 95 x 125 x 13	-	-	-	
11B	Freewheeling Ball Size 70 x 100 x 12	-	-	-	

* 65% Load for Items 1B through 5B
75% Load for Items 8B and 9B
100% Load for Items 6B and 7B
No calculations made for Items 10B or 11B

B

BEARING LOADS AND LIVES - 3000-HOUR TBO, TWIN 500 HP

ading * adial (lb)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
834	-	7,764	6,000	13	150,000
732	999	6,390	6,000	30	420,000
1,818	-	11,486	1,575	11	173,000
665	819	6,450	1,575	11	142,000
2,353	-	5,534	968	6	73,000
4,202	6,734	2,902	337	5	37,000
3,509	-	3,754	337	2	19,000
977	-	5,060	6,000	18	360,000
365	240	5,686	6,000	16	210,000
-	-	-	-	-	-
-	-	-	-	-	-

A

TABLE IV. SUMMARY OF FUNCTIONA
TWIN 500 HP, 3000- A

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Line Velocity (ft/min)	T
1G	Input Spiral Bevel Pinion	500	3,036	5.5	21	1.4	6,000	5,998	
2G	Input Spiral Bevel Gear	500	3,036	5.5	80	1.4	1,575	5,998	
3G	Sun Gear	667	4,100	8.29762	36	1.81	1,575	-	
4G	Planet Pinion								
	With Sun Gear	667	4,100	8.29762	46	1.78	968	1,404	
	With Ring Gear	667	4,100	8.70139	46	1.78	-	1,404	
5G	Ring Gear	667	4,100	8.70139	132	1.81	-	-	
6G	Tail Rotor Drive Pinion	108	626	5.53	21	.70	6,000	5,994	
7G	Accessory Drive Pinion	-	-	6.36	21	.35	6,000	5,184	

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13

MARY OF FUNCTIONAL GEAR DATA,
N 500 HP, 3000- AND 6000-HOUR TBO

	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in.) x 10^{-6}	Average Power Loss (hp)	Number of Meshes
0	5,998	259.0	26,945	177,250	8.1	2.189	2
5	5,998	259.0	27,026	177,283	8.1	2.189	2
5	-	93.2	38,941	160,902	16.64	.556	3
8	1,404	93.2	45,819	160,902	16.64	.556	-
	1,404	25.1	29,091	126,416	18.0	.418	-
	-	25.1	39,597	126,416	18.0	.418	2
0	5,994	187.0	17,349	123,050	9.7	.439	1
0	5,184	-	-	-	-	-	1

A

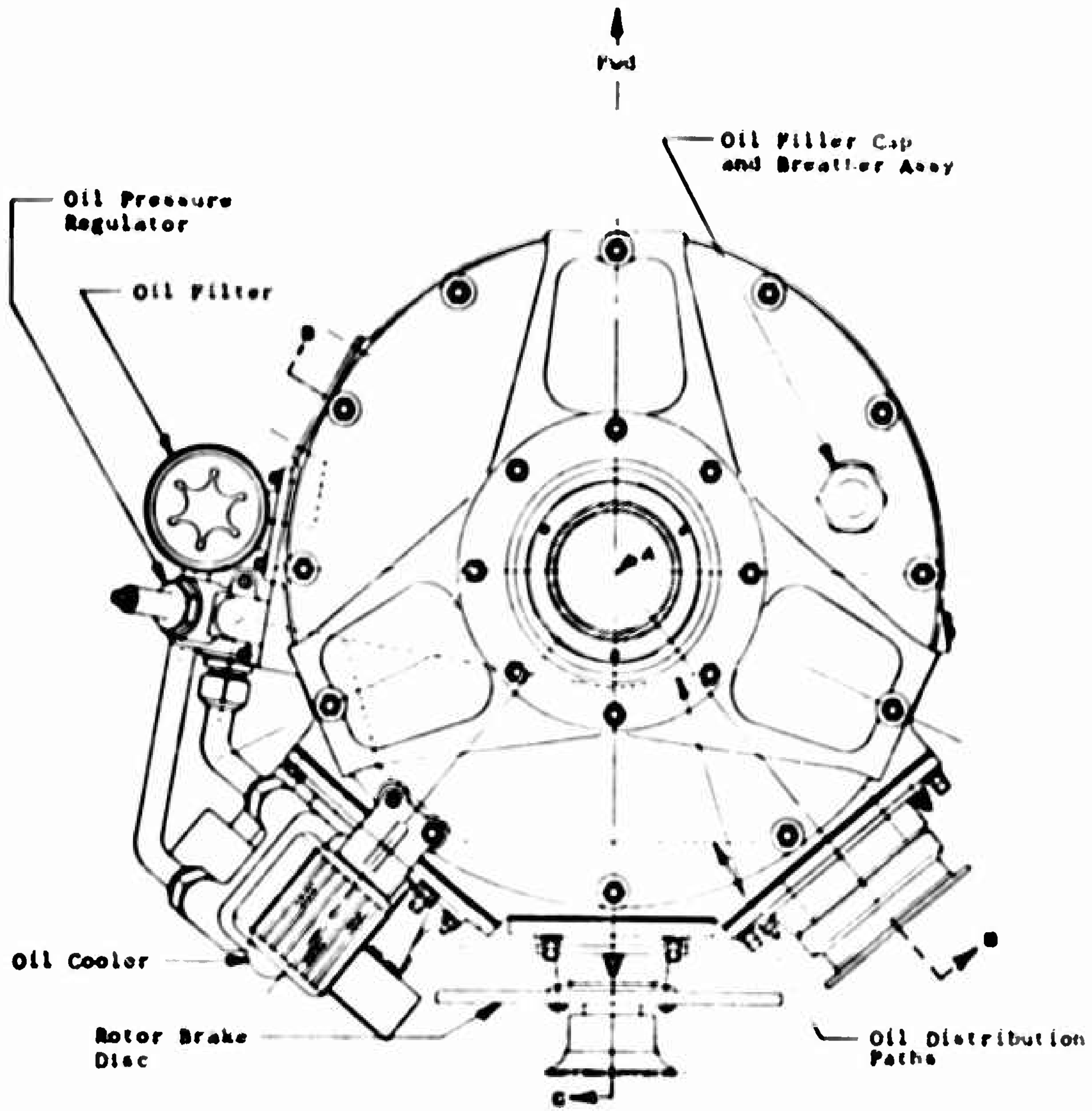

Figure 5. Twin 500-HP Transmission, 6000-Hour TBO.

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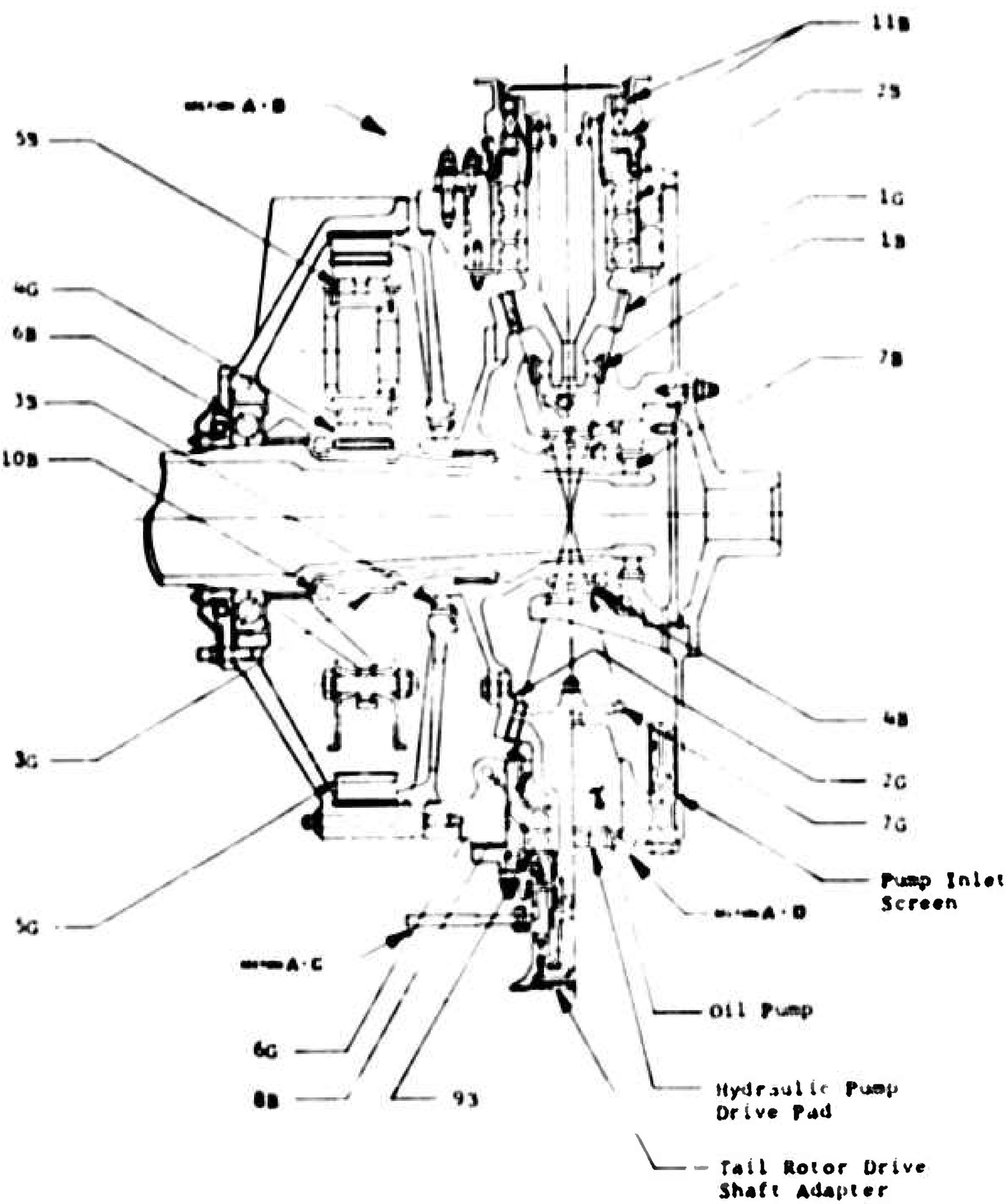
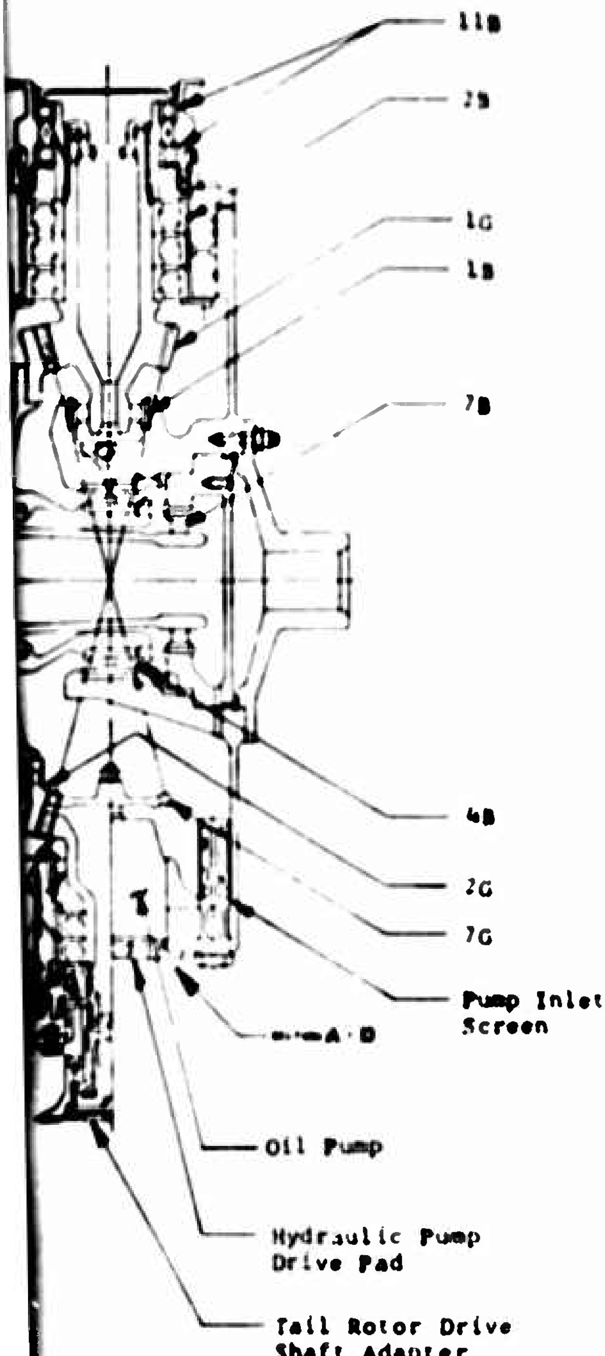

11B
2B
A-B
5B
1G
1B
4G
6B
7B
3B
10B
4B
3G
2G
7G
Pump Inlet
Screen
5G
A-D
A-C
Oil Pump
6G
8B
9B
Hydraulic Pump
Drive Pad
Tail Rotor Drive
Shaft Adapter



GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Sun Gear
4G	Planet Pinion
5G	Ring Gear
6G	Tail Rotor Drive Pinion
7G	Accessory Drive Pinion

BEARINGS

Item No.	Name
1B	Input Pinion Roller Size 304
2B	Input Pinion Ball Size 7212
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10
4B	Input Gear Ball Size 6918 With 13/32 Balls
5B	Planet Roller Size 11 x 11 x 3.86 P.D. 2 Rows: 16 Rollers/Row
6B	Mast Ball 204-040-136
7B	Mast Roller Size 1011
8B	Tail Rotor Pinion Roller Size 1911
9B	Tail Rotor Pinion Ball 204-040-143
10B	Planetary Support Ball Size 95 x 125 x 13
11B	Freewheeling Ball Size 70 x 100 x 12

4

TABLE V. SUMMARY OF BEARING LOADS AND LIVES -

Item	Bearing	Loading *			Bearing Hou[rs] Li[fe]
		Axial	Radial (lb)	Moment (in.-lb)	
1B	Input Pinion Roller Size 304	-	634	-	17[…]
2B	Input Pinion Ball Size 7212	1,287	904	1,087	11[…]
3B	Input Gear Roller Size 110 x 140; 32 - 8.5 x 10	-	1,636	-	17[…]
4B	Input Gear Ball Size 6918 with 13/32 Balls	1,175	703	1,032	16[…]
5B	Planet Roller 2 Rows; 16 Rollers/Row Size 11 x 11 x 3.86 PD	-	2,100	-	18[…]
6B	Mast Ball 204-040-136	5,370	4,202	6,734	11[…]
7B	Mast Roller Size 1011	-	3,509	-	15[…]
8B	Tail Rotor Pinion Roller Size 1911	-	918	-	21[…]
9B	Tail Rotor Pinion Ball 204-040-143	623	321	214	24[…]
10B	Planetary Support Ball Size 95 x 125 x 13	-	-	-	[illegible]
11B	Freewheeling Ball Size 70 x 100 x 12	-	-	-	[illegible]

* 65% Load for Items 1B through 5B
75% Load for Items 8B and 9B
100% Load for Items 6B and 7B
No calculations made for Items 10B or 11B

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13

G LOADS AND LIVES - 6000-HOUR TBO, TWIN 500 HP

Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
-	17,780	6,000	11	120,000
1,087	11,296	6,000	26	360,000
-	17,026	1,575	11	173,000
1,032	16,720	1,575	11	142,000
-	18,272	968	6	73,000
6,734	11,605	337	5	37,000
-	15,016	337	2	19,000
-	21,504	6,000	17	330,000
214	24,192	6,000	18	210,000
-	-	-	-	-
-	-	-	-	-

TWIN 1500-HP TRANSMISSION

The twin 1500-hp transmission comprises the total speed reduction from engines to main rotor mast. Each engine drives into the first-stage reduction, which is a spiral bevel gear set (Figure 6, items 1G and 2G). The bevel gear (Figure 6, item 2G) has a one-way overrunning clutch on the lower end which drives into a helical herringbone pinion. The helical pinions (Figure 6, item 3G) drive a "bull" helical herringbone gear (Figure 6, item 4G). The gear torque is split at the bull gear, where 1000 hp from each engine is transmitted to the main rotor mast and 375 hp total is transmitted to the accessories and tail rotor drive. The "bull" gear drives the lower planetary sun gear, which is splined to the bull gear shaft. The lower planetary stage consists of a sun gear (Figure 6, item 5G), four planetary idlers (Figure 6, item 6G) in a "ball joint carrier," and a floating ring gear (Figure 6, item 7G). The lower carrier drives the upper sun gear through a mating involute spline. The upper planetary assembly consists of a sun gear (Figure 6, item 8G), six planetary idlers (Figure 6, item 9G) in a rigid carrier, and a floating ring gear (Figure 6, item 10G). The upper carrier splines to and drives the main rotor mast (Figure 6, Tables VI and VII; Figure 7, Tables VII and IX; Figure 8, Table X).

The tail rotor drive system is comprised of a spur gear set (Figure 6, items 11G and 12G) which drives into a spiral bevel gear set (Figure 6, items 13G and 14G). The bevel gear has a spur gear (Figure 6, item 15G) mounted on the outboard end which drives the accessory gear system (Figure 6, items 16G and 17G).

The oil system consists of a constant-displacement 12.5-gpm pump driven off the tail rotor drive takeoff spur gear (Figure 6, item 11G), external lines and internal passages, oil jets, and oil manifold sufficient to provide cooling and lubrication for all gears and bearings. Oil jets are provided for intermittent lubrication of planetary gears (by virtue of planetary rotation past the jets) and oil into and out of mesh on the bevel and helical gears. Bearings are lubricated by oil flowing from passages in the housing into annuli around the bearings, and finally into the bearings via ground slots between the bearings, or simply by jets into the bearings. External lines are used to carry the oil from the pump to a 10-micron full-flow filter, then to the oil cooler, and finally to the transmission oil manifold. The oil filter, cooler, and cooling fan will be mounted on the transmission and hence require no flexible hoses to accommodate relative motion between airframe

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and transmission. The oil cooling fan is mounted on the accessory drive and remains operational in the event of single-engine operation.

The transmission cases are fabricated from cast magnesium with the exception of the aluminum spacer case, which houses the planetary ring gear and the support case which attaches to the airframe. The method of mounting the ring gear provides radial compliance for the ring gear, and hence relative motion between gear and spline occurs as the planets orbit inside the ring gear. The relative motion with associated wear potential precludes the use of magnesium since magnesium exhibits little wear resistance for this application. The use of aluminum alloy, however, completely eliminates this wear problem.

Casting geometry and volume are necessarily the same for each TBO period designs for comparable transmission cases made from aluminum and magnesium. The wall thicknesses, transition geometry, and structural proportioning are as required to provide adequate dimensional consistency, radiographic quality, and gear mounting rigidity, respectively. The difference in modulii of elasticity is considered insufficient to allow section changes in an attempt to adjust slight rigidity changes from magnesium to aluminum, especially in complex thin wall castings.

TWIN 4800-HP TRANSMISSION

The twin 4800-hp engines drive the main transmission through identical engine reduction gearboxes, which are mounted on opposite sides of the transmission main case. Engine output speed is reduced through this engine reduction gearbox by a 67:99 ratio spiral bevel gear set (Figure 9, items 1G and 2G). This gear set drives the main spiral bevel input pinion through an overrunning clutch. The main input pinions (Figure 9, item 3G) independently drive the "bull" spiral bevel gear (Figure 9, item 4G(A)). The combined twin-engine power is then transmitted from this point to the main rotor through two fixed-ring, sun-input/carrier-output spur gear planetary drives of 3.875:1 ratio each (Figure 9, Tables XI and XII; Figure 10, Tables XIII and XIV; Figure 11, Table XV.

The shaft of the main input bevel gear (Figure 9, item 4G(A)) is splined to the sun gear of the lower planetary stage. The lower sun gear (Figure 9, item 5G) drives four planet idler gears inside a fixed ring gear (Figure 9, item 7G). The gear loads are transmitted from the planet idlers (Figure 9, item 6G) to the carrier plates, which in turn transmit the loads to

the attaching bolts of the spherical ball bearings in the spider. The spider has four legs equally spaced in which the spherical ball bearings are fitted. The spider drives the upper planetary sun gear (Figure 9, item 8G) through an involute spline.

The upper planetary sun gear drives six planetary idler gears (Figure 9, item 9G) inside a fixed ring gear (Figure 9, item 10G). The idler gears are mounted in a rigid carrier which transmits power to the main rotor mast through an involute spline.

The tail rotor and accessory drive power is split from the main spiral bevel gear (Figure 9, Item 4G(A)) by a spiral bevel pinion (Figure 9, item 11G) at a 90° shaft angle. The tail rotor shafting is driven by using a curvic face coupling splined to the pinion.

The accessories are driven by a spur gear (Figure 9, item 12G) mounted on and driven by the spiral bevel pinion (Figure 9, item 11G). The accessory drive consists of two 56/48 tooth spur clusters (Figure 9, items 13G(A) and 13G(B)), driven by the accessory spur gear, for the twin generator drives. Each spur pinion drives a spur gear (Figure 9, item 14G) for the hydraulic pump drives.

The oil system components consist of a 20-gallon-capacity gravity return and externally finned wet sump, a 70-gpm oil pump driven by an offset spur gear (Figure 9, item 15G) on the lower end of the main bevel gear shaft, a two-phase filter system consisting of a coarse filter screen and a 10-micron filter, an external oil cooler, a bypass-type pressure limiting valve, internal oil passages, externally removable oil jets, and associated tubing, fittings, and monitoring devices.

The oil is drawn into the pump through the coarse filter screen directly from the sump and routed out the front of the transmission main case to the oil cooler and the 10-micron filter and then back to the transmission main case supply manifold. A system of internal oil passages in the main case carries oil to the bearings and jets, which sprays gear meshes and lower case roller bearings. Jet sprays, fed by rigid external oil lines, supply lubrication to the top case bearings and planetary meshes and bearings. Sight glasses are provided in the main case and sump for high and low oil level observation.

The transmission upper case houses the main rotor mast thrust bearing, which transfers lift from the rotating main rotor to the upper case. The upper case is attached to an intermediate

case, which has three external cast-on lugs for mounting the transmission assembly and transmitting lift to the airframe. The main case houses the lower main rotor mast bearing, which, with the upper bearing, provides a moment restraint to transmit radial (bending) mast loads through the transmission housing and into the airframe.

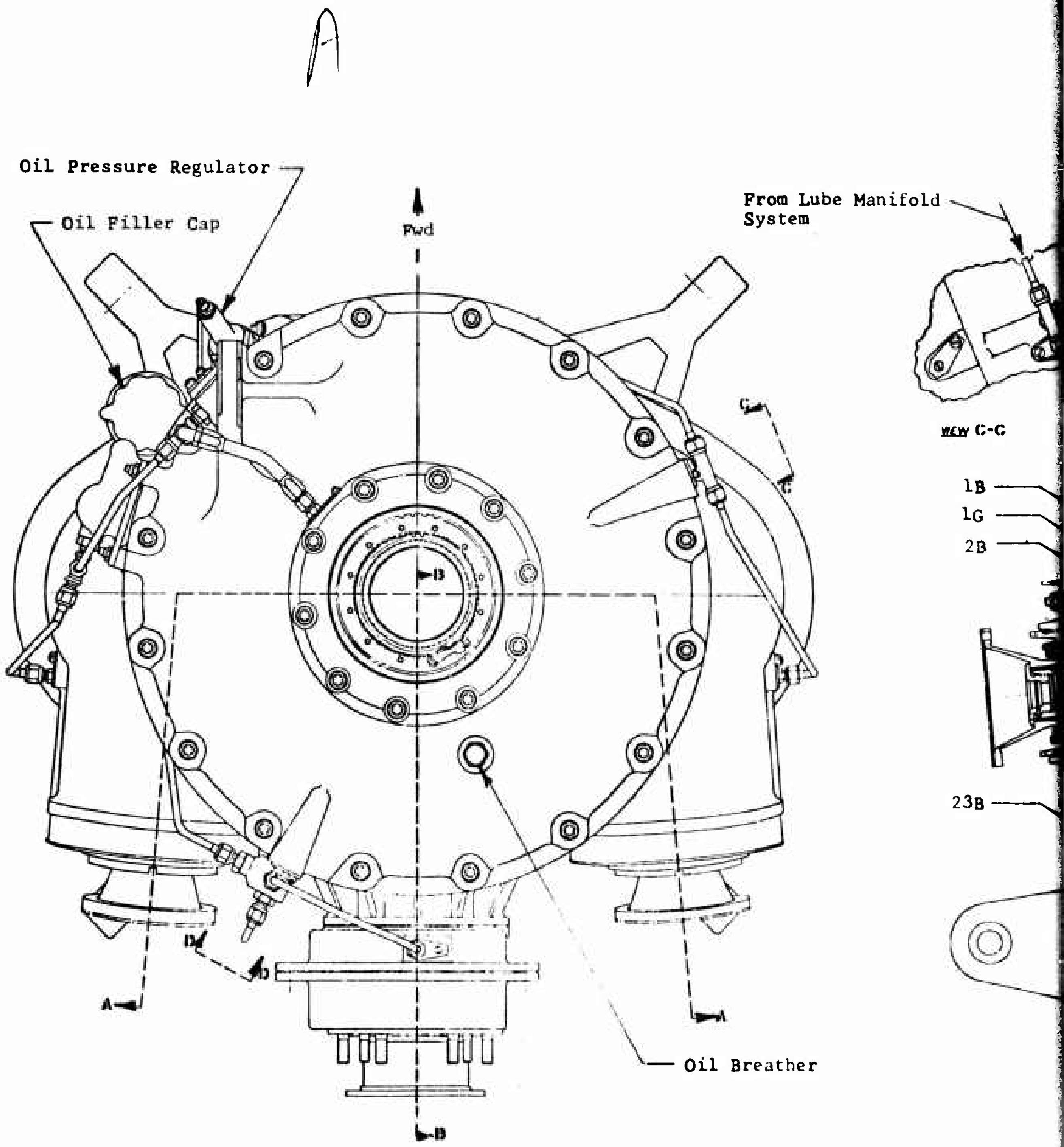
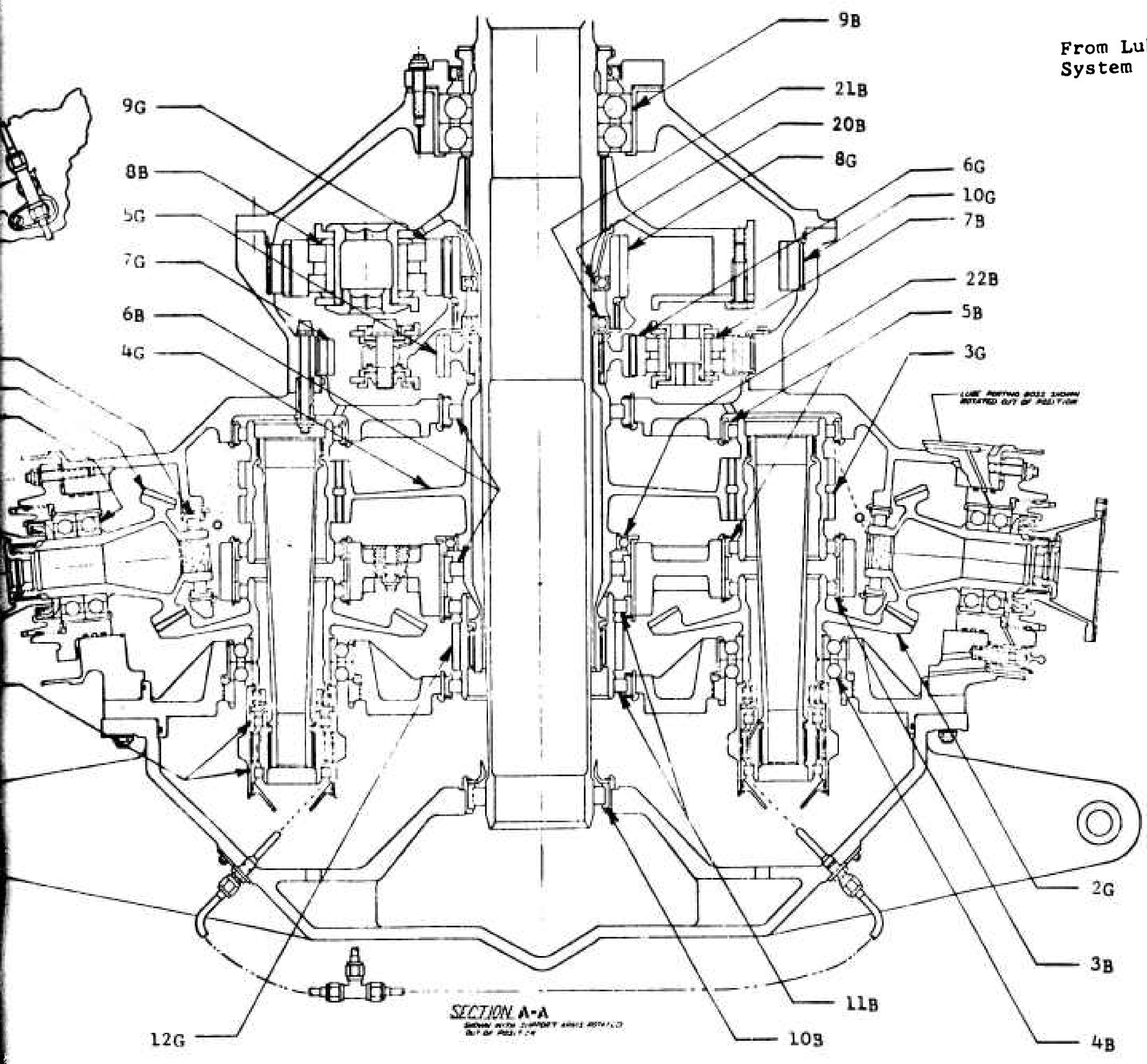


Figure 6. Twin 1500-HP Transmission, 1200-Hour TBO.

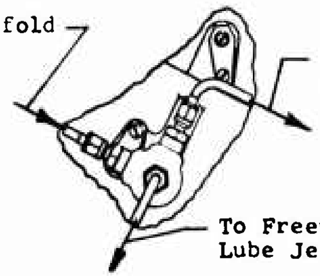
13

From Lube System



SECTION A-A

C



VIEW D-D

GEARS

Item No.	Name
1G	Input Spiral Bevel
2G	Input Spiral Bevel
3G	Herringbone Pinion
4G	Herringbone Gear
5G	Lower Sun Gear
6G	Lower Planet Pinio
7G	Lower Ring Gear
8G	Upper Sun Gear
9G	Upper Planet Pinio
10G	Upper Ring Gear
11G	Offset Spur Pinion
12G	Offset Spur Gear
13G	Sump Bevel Pinion
14G	Sump Bevel Gear
15G	Accessory Spur Pin
16G	Accessory Spur Idl
17G	Accessory Spur Gea

BEARINGS

Item No.	Name
1B	Input Pinion Rolle Size 207
2B	Input Pinion Ball Size 7210
3B	Input Gear Roller Size 1914
4B	Input Gear Ball Size 7014

GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Herringbone Pinion
4G	Herringbone Gear
5G	Lower Sun Gear
6G	Lower Planet Pinion
7G	Lower Ring Gear
8G	Upper Sun Gear
9G	Upper Planet Pinion
10G	Upper Ring Gear
11G	Offset Spur Pinion
12G	Offset Spur Gear
13G	Sump Bevel Pinion
14G	Sump Bevel Gear
15G	Accessory Spur Pinion
16G	Accessory Spur Idler
17G	Accessory Spur Gear

BEARINGS

Item No.	Name
1B	Input Pinion Roller Size 207
2B	Input Pinion Ball Size 7210
3B	Input Gear Roller Size 1914
4B	Input Gear Ball Size 7014

BEARINGS - Continued

Item No.	Name
5B	Herringbone Pinion Roller Size 1914
6B	Herringbone Gear Roller 204-040-271
7B	Lower Planet Roller 204-040-725/-132
8B	Upper Planet Roller 563-040-112/-113
9B	Mast Ball 205-040-165
10B	Mast Roller 563-040-270
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11
12B	35T Spur Pinion Roller Size 206; 13 - 9 x 11
13B	Pinion Roller Size 208
14B	Pinion Ball Size 7212
15B	Gear Roller Size 211
16B	Gear Ball Size 7210
17B	34T Acc. Drive Roller Size 1004
18B	39T Acc. Drive Idler Roller Size 1905
19B	34T Acc. Drive (Single Mesh) Roller Size 1906
20B	Upper Planetary Support Ball 563-040-120
21B	Lower Planetary Support Ball 563-040-120
22B	Herringbone Gear Support Ball Size 140 x 165
23B	Freewheeling Roller Size 1009

A

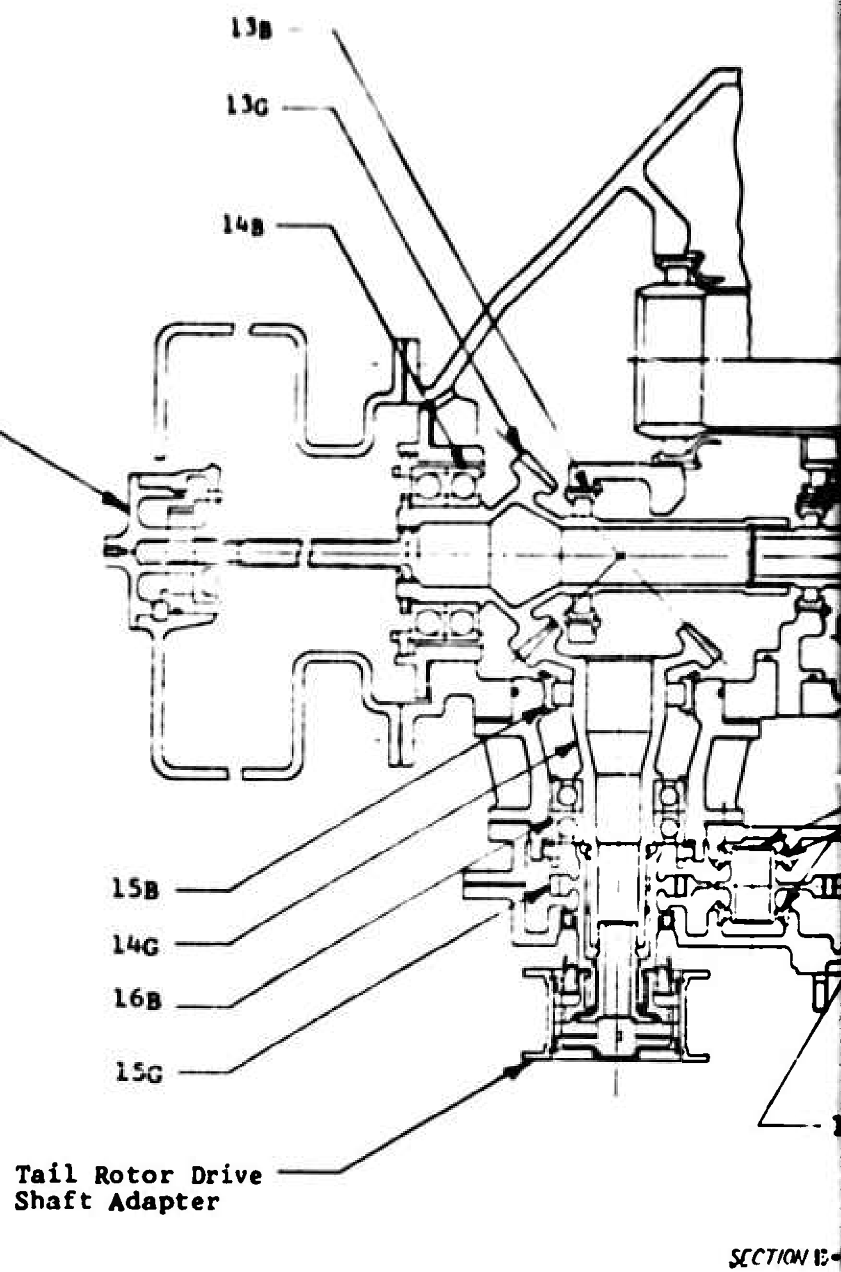


Figure 6 - Continued.

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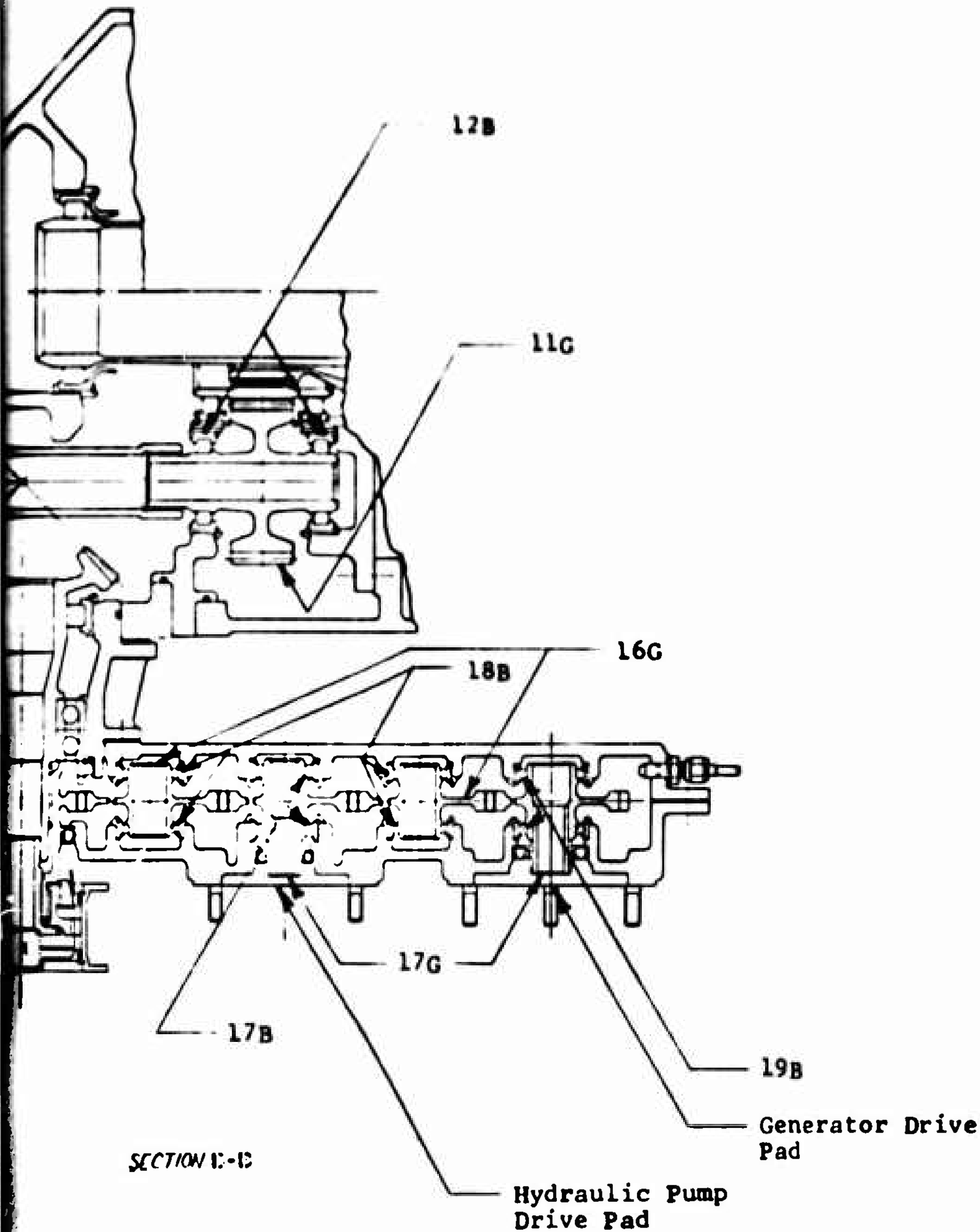
12B
11G
16G
18B
17G
17B
19B
Generator Drive Pad
Hydraulic Pump Drive Pad
SECTION E-E

1

TABLE VI. SUMMARY OF BEARING LOADS AND LIVES - 1200-HO

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bearing Li Hours L10 *
1B	Input Pinion Roller Size 207	-	916	-	9,080
2B	Input Pinion Ball Size 7210	372	326	125	2,844
3B	Input Gear Roller Size 1914	-	605	-	98,304
4B	Input Gear Ball Size 7014	490	648	329	3,311
5B	Herringbone Pinion Roller Size 1914	-	1,652	-	2,980
6B	Herringbone Gear Roller 204-040-271	-	1,652	-	32,964
7B	Lower Planet Roller 204-040-725/-132	-	2,025	-	4,592 (M-50)
8B	Upper Planet Roller 563-040-112/-113	-	4,660	-	2,684
9B	Mast Ball 205-040-165	5,176	2,036	3,272	10,588
10B	Mast Roller 563-040-270	-	2,668	-	46,037
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11	-	1,125	-	71,591
12B	35T Spur Pinion Roller Size 206; 13 - 9 x 11	-	1,125	-	2,974
13B	Pinion Roller Size 208	-	1,520	-	5,934

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G LOADS AND LIVES - 1200-HOUR TBO, TWIN 1500 HP

g * l)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
16	-	9,080	24,000	43	840,000
26	125	2,844	24,000	58	1,200,000
05	-	98,304	11,871	37	830,970
48	329	3,311	11,871	41	830,970
52	-	2,980	11,871	35	830,970
52	-	32,964	3,011	22	361,320
25	-	4,592 (M-50)	3,743	11	131,005
50	-	2,684	808	5	44,440
36	3,272	10,588	252	14	27,720
58	-	46,037	252	3	24,192
25	-	71,591	3,011	7	361,320
25	-	2,974	4,387	15	131,610
20	-	5,934	4,387	6	175,480

/1

TABLE VI - Continued

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bearing Hours L10 *
14B	Pinion Ball Size 7212	1,643	827	800	3,278
15B	Gear Roller Size 211	-	3,407	-	3,398
16B	Gear Ball Size 7210	504	670	285	4,336
17B	34T Accessory Drive Roller Size 1004	-	498	-	4,822
18B	39T Accessory Drive Idler Roller Size 1905	-	498	-	2,916
19B	34T Accessory Drive (Single Mesh) Roller Size 1906	-	314	-	17,562
20B	Upper Planetary Support Ball 563-040-120	-	-	-	3,280,000
21B	Lower Planetary Support Ball 563-040-120	-	-	-	512,260
22B	Herringbone Gear Support Ball Size 140 x 165	-	-	-	91,645
23B	Freewheeling Roller Size 1009	-	-	-	-

* 65% - Items 1 through 10
75% - Items 11 through 19
100% - Items 20 through 23

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C

TABLE VI - Continued

Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
800	3,278	4,387	8	263,220
-	3,338	4,150	23	228,250
285	4,336	4,150	7	207,500
-	4,822	4,150	20	83,000
-	2,916	3,618	10	90,450
-	17,562	4,150	22	124,500
-	3,280,000	723	13	57,840
-	512,260	2,036	15	210,770
-	91,645	3,011	15	210,770
-	-	-	-	-

A

TABLE VII. SUMMARY OF FUNCTIONAL GEAR DATA -

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Line Velocity (ft/min)
1G	Input Spiral Bevel Pinion	1,500	1,926	10.000	46	1.200	24,000	28,903
2G	Input Spiral Bevel Gear	1,500	1,926	10.000	93	1.200	11,871	28,903
3G	Herringbone Pinion	1,500	4,631	10.176	35	.906	11,871	10,689
4G	Herringbone Gear	1,500	4,631	10.176	138	.906	3,011	10,689
5G	Lower Sun Gear	2,000	3,121	8.500	57	1.450	3,011	-
6G	Lower Planet Pinion							
	With Sun Gear	2,000	3,121	8.500	31	1.430	3,743	3,574
	With Ring Gear	2,000	3,121	8.500	31	1.430	3,743	3,574
7G	Lower Ring Gear	2,000	3,121	8.500	119	1.200	-	-
8G	Upper Sun Gear	2,000	6,998	7.796	48	2.26	975	-
9G	Upper Planet Pinion							
	With Sun Gear	2,000	6,998	7.796	43	2.26	808	1,165
	With Ring Gear	2,000	7,305	8.138	43	2.26	808	1,118
10G	Upper Ring Gear	2,000	7,305	8.138	138	1.737	-	-
11G	Offset Spur Pinion	375	2,760	8.958	35	1.600	4,387	4,487

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13

IONAL GEAR DATA - 1200-HOUR TBO, TWIN 1500 HP							
PM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
,000	28,903	168	24,327	156,700	750	3.975	2
,871	28,903	168	24,429	156,700	750	3.975	2
,871	10,689	81	52,053	121,964	95	.866	2
,011	10,689	81	46,139	121,964	95	.866	2
,011	-	83	40,606	169,699	24	.695	4
,743	3,574	83	44,902	169,699	24	.695	-
,743	3,574	28	45,481	169,224	29	.462	-
-	-	28	47,259	169,224	29	.462	4
975	-	74	49,928	169,940	15	.662	6
808	1,165	74	56,993	169,940	15	.662	-
808	1,118	33	54,329	169,994	16	.636	-
-	-	33	71,333	169,994	16	.636	6
,387	4,487	86	37,421	146,799	31	.661	1

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TABLE VII - Co

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch I Veloci (ft/mi
12G	Offset Spur Gear	375	2,760	8.958	51	1.600	3,011	4,407
13G	Sump Bevel Pinion	375	2,580	7.000	35	1.200	4,387	5,743
14G	Sump Bevel Gear	375	2,580	7.000	37	1.200	4,150	5,743
15G	Accessory Spur Pinion	100	895	10.000	34	.520	4,150	3,694
16G	Accessory Spur Idler	100	895	10.000	39	.520	3,618	3,694
17G	Accessory Spur Gear	100	895	10.000	34	.520	4,150	3,694

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B

ABLE VII - Continued

RPM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
,011	4,487	86	37,421	146,799	31	.661	1
,387	5,743	91	32,600	203,600	750	1.185	1
,150	5,743	91	32,700	203,600	750	1.185	1
,150	3,694	89	40,513	166,981	25	.229	1
,618	3,694	89	39,716	166,981	25	.229	2
,150	3,694	89	40,513	166,981	25	.229	1

A

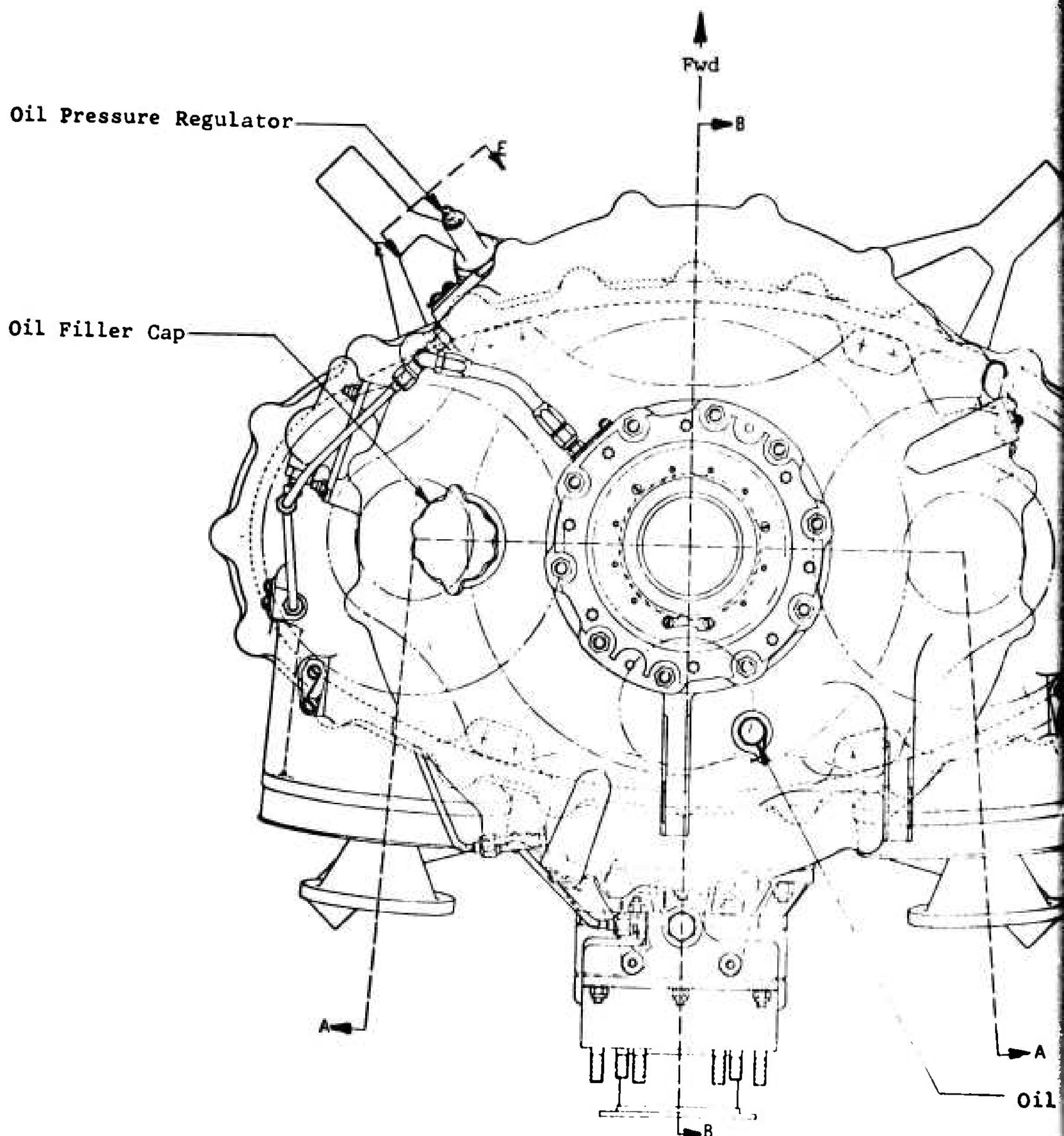
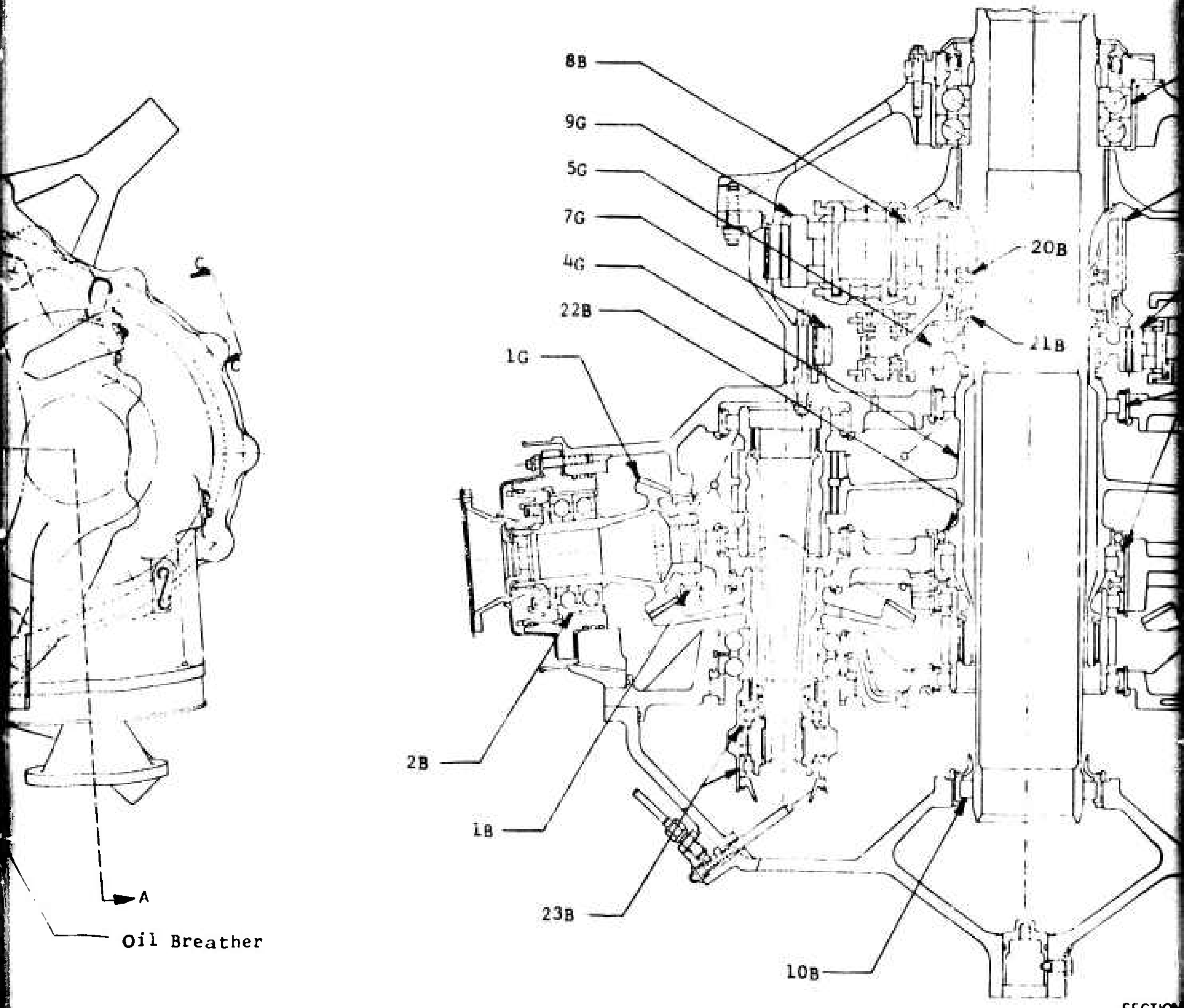


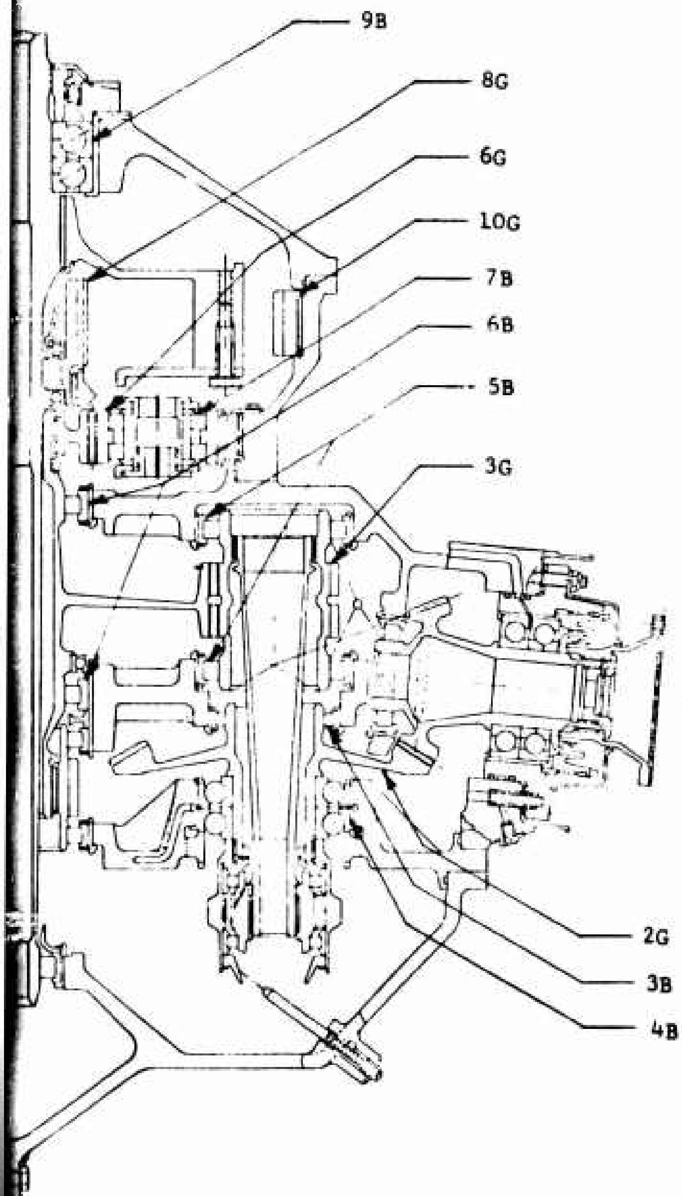
Figure 7. Twin 1500-HP Transmission, 3000-Hour T30.

B



8B
9G
5G
7G
4G
22B
1G
20B
21B
2B
1B
23B
10B
A
Oil Breather
SECTION

C



SECTION A-A

GEARS

Item No.	Name
1G	Input Spiral
2G	Input Spiral
3G	Herringbone
4G	Herringbone
5G	Lower Sun G
6G	Lower Plane
7G	Lower Ring
8G	Upper Sun G
9G	Upper Plane
10G	Upper Ring
11G	Offset Spur
12G	Offset Spur
13G	Sump Bevel
14G	Sump Bevel
15G	Accessory S
16G	Accessory S
17G	Accessory S

BEARINGS

Item No.	Name
1B	Input Pinio Size 208
2B	Input Pinio Size 7211
3B	Input Gear Size 1913
4B	Input Gear Size 7212

D

GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Herringbone Pinion
4G	Herringbone Gear
5G	Lower Sun Gear
6G	Lower Planet Pinion
7G	Lower Ring Gear
8G	Upper Sun Gear
9G	Upper Planet Pinion
10G	Upper Ring Gear
11G	Offset Spur Pinion
12G	Offset Spur Gear
13G	Sump Bevel Pinion
14G	Sump Bevel Gear
15G	Accessory Spur Pinion
16G	Accessory Spur Idler
17G	Accessory Spur Gear

BEARINGS

Item No.	Name
1B	Input Pinion Roller Size 208
2B	Input Pinion Ball Size 7211
3B	Input Gear Roller Size 1913
4B	Input Gear Ball Size 7212

BEARINGS - Continued

Item No.	Name
5B	Herringbone Pinion Roller Size 1014
6B	Herringbone Gear Roller 204-040-271
7B	Lower Planet Roller 204-040-725/-132
8B	Upper Planet Roller 563-040-112/-113
9B	Mast Ball 205-040-165
10B	Mast Roller 563-040-270
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11
12B	35T Spur Gear Roller Size 206; 13 - 9 x 11
13B	Pinion Roller Size 208
14B	Pinion Ball Size 7211
15B	Gear Roller Size 1016
16B	Gear Ball Size 7211
17B	34T Acc. Drive Roller Size 1005
18B	39T Acc. Idler Roller Size 1005
19B	34T Acc. Drive (Single Mesh) Roller Size 1005
20B	Upper Planetary Support Ball 563-040-120
21B	Lower Planetary Support Ball 563-040-120
22B	Herringbone Gear Support Ball Size 140 x 165
23B	Freewheeling Roller Size 1009

A

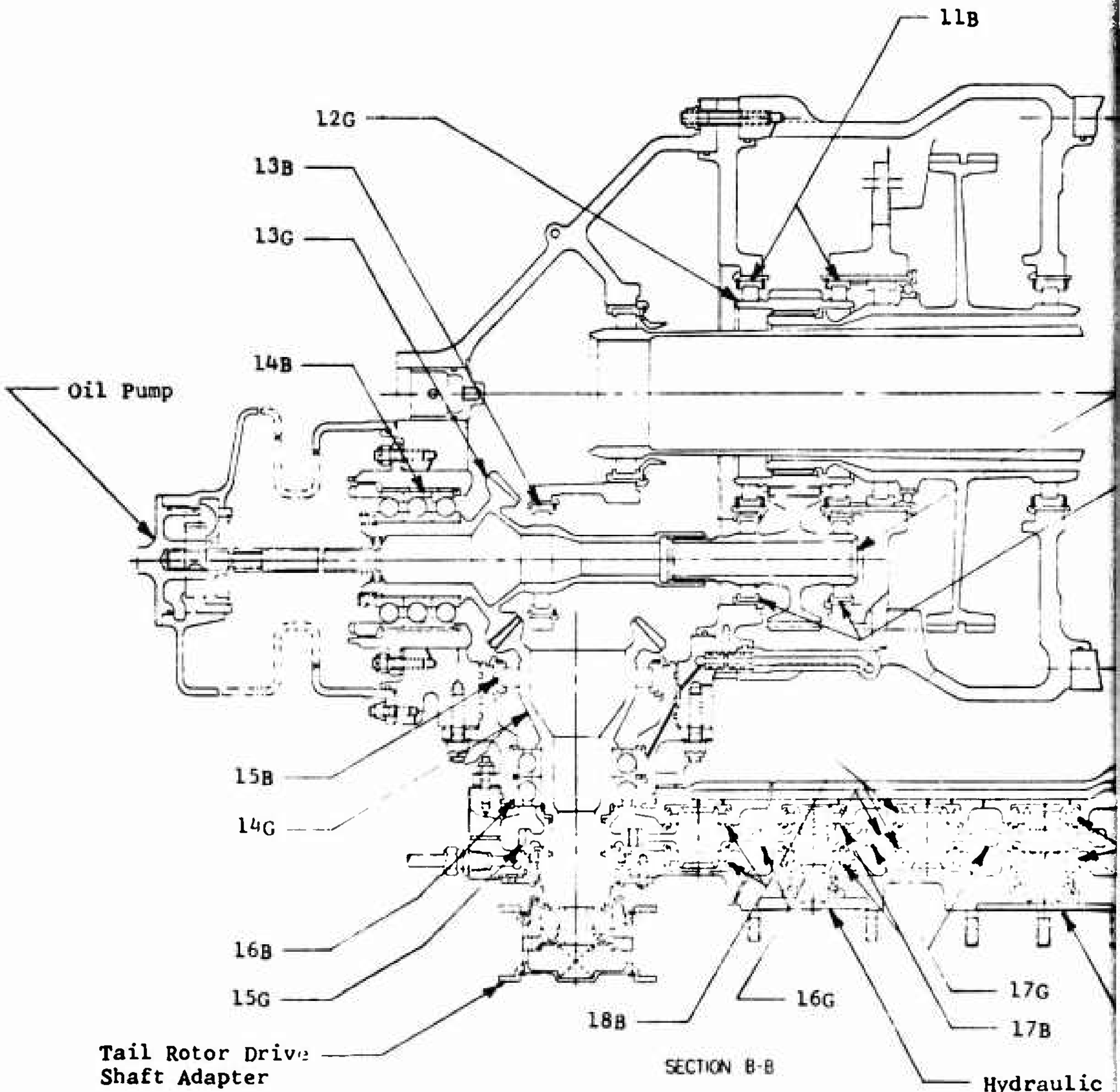


Figure 7 - Continued.

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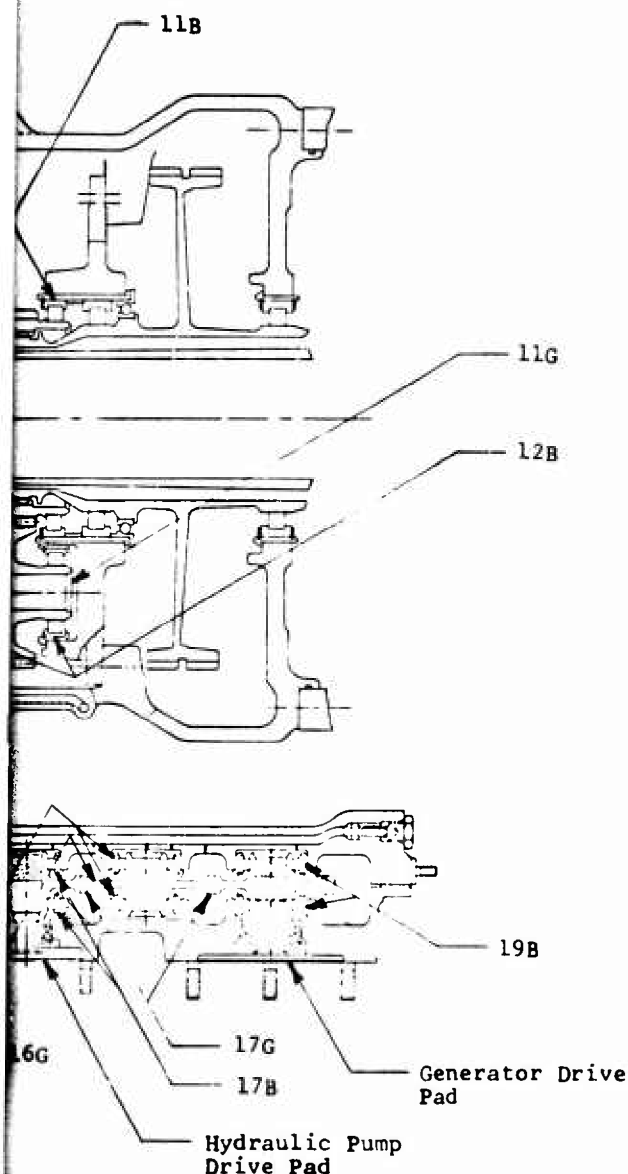
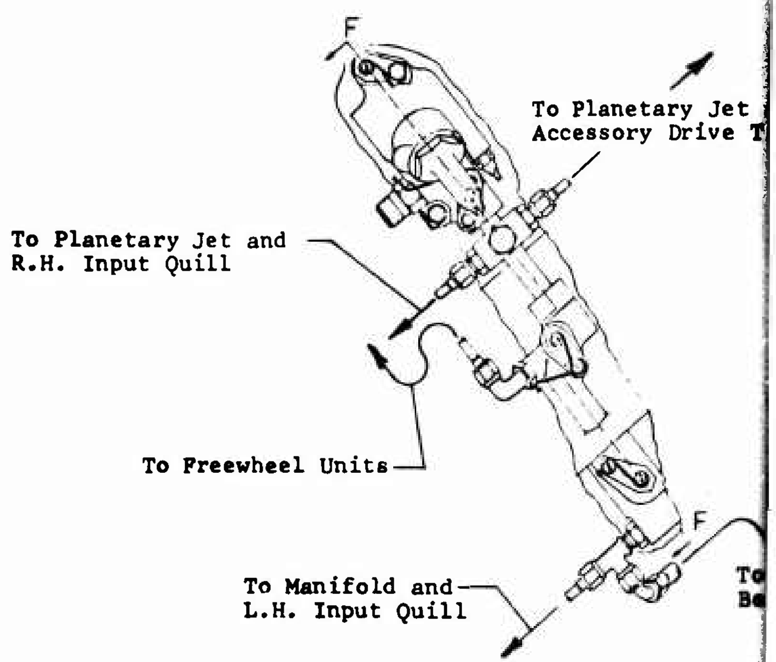
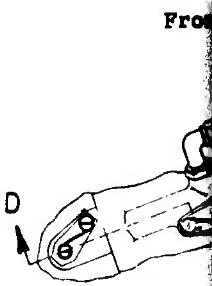
Pj


11B
11G
12B
19B
6G
17G
17B
Generator Drive Pad
Hydraulic Pump Drive Pad

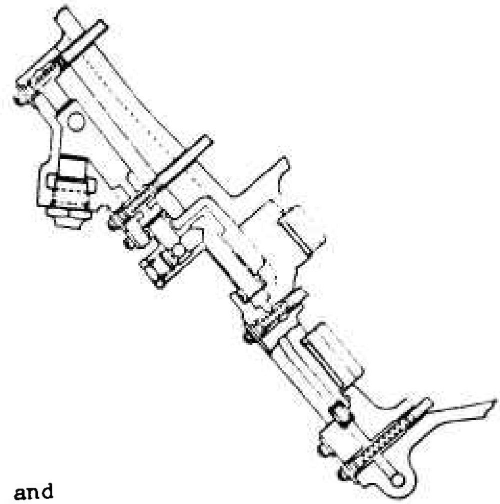

F
To Planetary Jet
Accessory Drive T
To Planetary Jet and R.H. Input Quill
To Freewheel Units
F
To Manifold and L.H. Input Quill

VIEW E-E

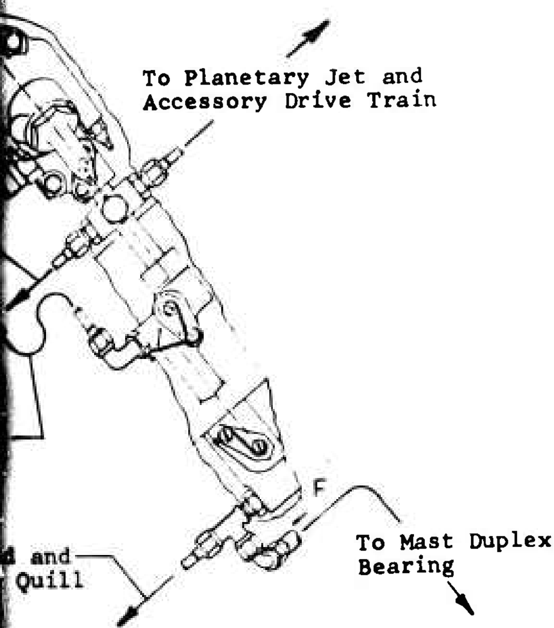
C



VIEW

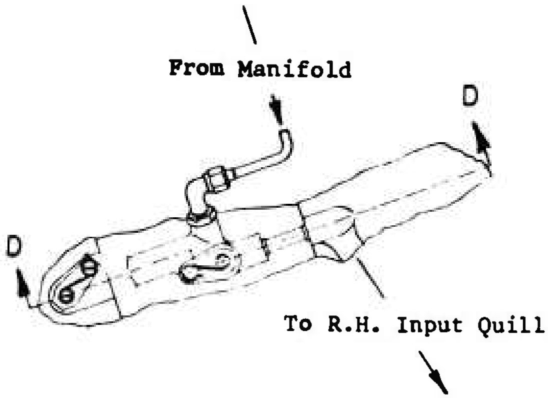


SECTION F-F

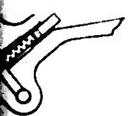


VIEW E-E

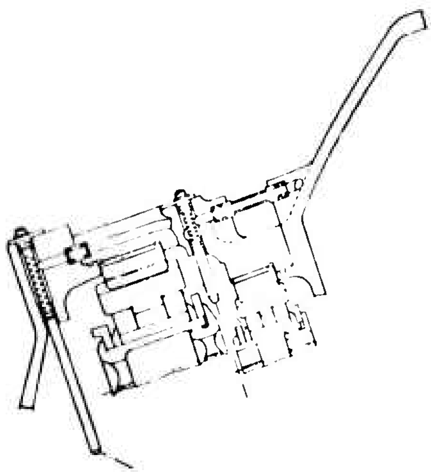
D



VIEW C-C



F-F



SECTION D-D

A

TABLE VIII. SUMMARY OF BEARING LOADS AND LIVES -

Item	Bearing	Loading * Axial	Radial (lb)	Moment (in.-lb)	Bearin Hou L1
1B	Input Pinion Roller Size 208	-	913	-	11,
2B	Input Pinion Ball Size 7211	371	321	140	6,
3B	Input Gear Roller Size 1913	-	604	-	29,
4B	Input Gear Ball Size 7212	620	621	299	6,
5B	Herringbone Pinion Roller Size 1014	-	1,660	-	8,
6B	Herringbone Gear Roller 204-040-721	-	1,649	-	32,
7B	Lower Planet Roller 024-040-725/-132	-	2,269	-	10,
8B	Upper Planet Roller 563-040-112/-113	-	10,427	-	6,
9B	Mast Ball 205-040-165	5,176	2,036	3,272	10,
10B	Mast Roller 563-040-270	-	2,668	-	46,
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11	-	1,125	-	71,
12B	35T Spur Gear Roller Size 206; 13 -19 x 11	-	1,125	-	5,
13B	Pinion Roller Size 208	-	1,190	-	14,

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13

EARING LOADS AND LIVES - 3000-HOUR TBO, TWIN 1500 HP

ding * dial lb)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
913	-	11,824	24,000	47	960,000
321	140	6,896	24,000	65	1,320,000
604	-	29,072	11,871	33	771,615
621	299	6,496	11,871	42	712,260
,660	-	8,583	11,871	38	830,970
,649	-	32,964	3,011	22	361,320
,269	-	10,128	3,743	11	121,005
,427	-	6,048	808	5	44,440
,036	3,272	10,588	252	4	27,720
,668	-	46,037	252	3	24,192
,125	-	71,591	3,011	22	361,320
,125	-	5,870	4,387	10	131,610
,190	-	14,870	4,387	13	175,480

A

TABLE VIII - Cont

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bear H L
14B	Pinion Ball Size 7211	846	636	478	[illegible]
15B	Gear Roller Size 1016	-	3,427	-	8
16B	Gear Ball Size 7211	525	687	317	6
17B	34T Accessory Drive Roller Size 1005	-	463	-	5
18B	39T Accessory Idler Roller Size 1005	-	463	-	6
19B	34T Accessory Drive (Single Mesh) Roller Size 1005	-	314	-	2
20B	Upper Planetary Support Ball 563-040-120	95	-	-	3,28
21B	Lower Planetary Support Ball 563-040-120	129	-	-	51
22B	Herringbone Gear Support Ball Size 140 x 165	169	-	-	9
23B	Freewheeling Roller Size 1009	-	-	-	

* 65% - Items 1 through 10
7[illegible] - Items 11 through 19
100% - Items 20 through 23

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B

TABLE VIII - Continued

ding * dial lb)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
636	478	5,170	4,387	18	241,285
,427	-	8,968	4,150	20	332,000
687	317	6,774	4,150	17.5	228,250
463	-	5,508	4,150	7.5	103,750
463	-	6,324	3,618	6.8	90,450
314	-	21,138	4,150	7.7	103,750
-	-	3,280,000	723	6	57,840
-	-	512,260	2,036	8	162,880
-	-	91,645	2,011	12	210,770
-	-	-	-	-	-

A

TABLE IX. SUMMARY OF FUNCT
TWIN 1500 HP, 30

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Line Velocity (ft/min)
1G	Input Spiral Bevel Pinion	1,500	1,926	10.000	46	1.200	24,000	28,903
2G	Input Spiral Bevel Gear	1,500	1,926	10.000	93	1.200	11,871	28,903
3G	Herringbone Pinion	1,500	4,631	10.176	35	.906	11,871	10,689
4G	Herringbone Gear	1,500	4,631	10.176	138	.906	3,011	10,689
5G	Lower Sun Gear	2,000	3,121	8.500	57	1.596	3,011	-
6G	Lower Planet Pinion							
	With Sun Gear	2,000	3,121	8.500	31	1.596	3,743	3,574
	With Ring Gear	2,000	3,121	8.500	31	1.596	3,743	3,574
7G	Lower Ring Gear	2,000	3,121	8.500	119	1.356	-	-
8G	Upper Sun Gear	2,000	6,998	7.796	48	2.529	975	-
9G	Upper Planet Pinion							
	With Sun Gear	2,000	6,998	7.796	43	2.529	808	1,165
	With Ring Gear	2,000	7,305	8.138	43	2.529	808	1,118
10G	Upper Ring Gear	2,000	7,305	8.138	138	1.944	-	-
11G	Offset Spur Pinion	375	2,760	8.958	35	1.600	4,387	4,487

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B

. SUMMARY OF FUNCTIONAL GEAR DATA,
TWIN 1500 HP, 3000- AND 6000-HOUR TBO

RPM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
24,000	28,903	168	24,327	156,700	750	3.975	2
11,871	28,903	168	24,429	156,700	750	3.975	2
11,871	10,689	81	52,053	121,964	95	.866	2
3,011	10,689	81	46,139	121,964	95	.866	2
3,011	-	80	36,890	160,631	25	.709	4
3,743	3,574	80	40,232	160,631	25	.709	-
3,743	3,574	26	40,742	160,630	28	.469	-
-	-	26	41,822	160,630	28	.469	4
975	-	70	44,618	160,647	15	.671	6
808	1,165	70	50,878	160,647	15	.671	-
808	1,118	32	49,026	160,642	16	.655	-
-	-	32	63,719	160,642	16	.655	6
4,387	4,487	86	37,421	146,799	31	.661	1

A

							T
Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	
12G	Offset Spur Gear	375	2,760	3.958	51	1.600	3
13G	Sump Bevel Pinion	375	2,580	7.000	35	1.200	4
14G	Sump Bevel Gear	375	2,580	7.000	37	1.200	4
15G	Accessory Spur Pinion	100	895	10.000	34	.520	4
16G	Accessory Spur Idler	100	895	10.000	39	.520	3
17G	Accessory Spur Gear	100	895	10.000	34	.520	4

A

TABLE IX - Conti

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Line Velocity (ft/min)
12G	Offset Spur Gear	375	2,760	8.958	51	1.600	3,011	4,487
13G	Sump Bevel Pinion	375	2,580	7.000	35	1.200	4,387	5,743
14G	Sump Bevel Gear	375	2,580	7.000	37	1.200	4,150	5,743
15G	Accessory Spur Pinion	100	895	10.000	34	.520	4,150	3,694
16G	Accessory Spur Idler	100	895	10.000	39	.520	3,618	3,694
17G	Accessory Spur Gear	100	895	10.000	34	.520	4,150	3,694

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B

TABLE IX - Continued

RPM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
3,011	4,487	86	31,941	146,799	31	.661	1
4,387	5,743	91	32,600	203,600	750	1.185	1
4,150	5,743	91	32,700	203,600	750	1.185	1
4,150	3,694	89	40,513	166,981	25	.229	1
3,618	3,694	89	39,716	166,981	25	.229	2
4,150	3,694	89	40,513	166,981	25	.229	1

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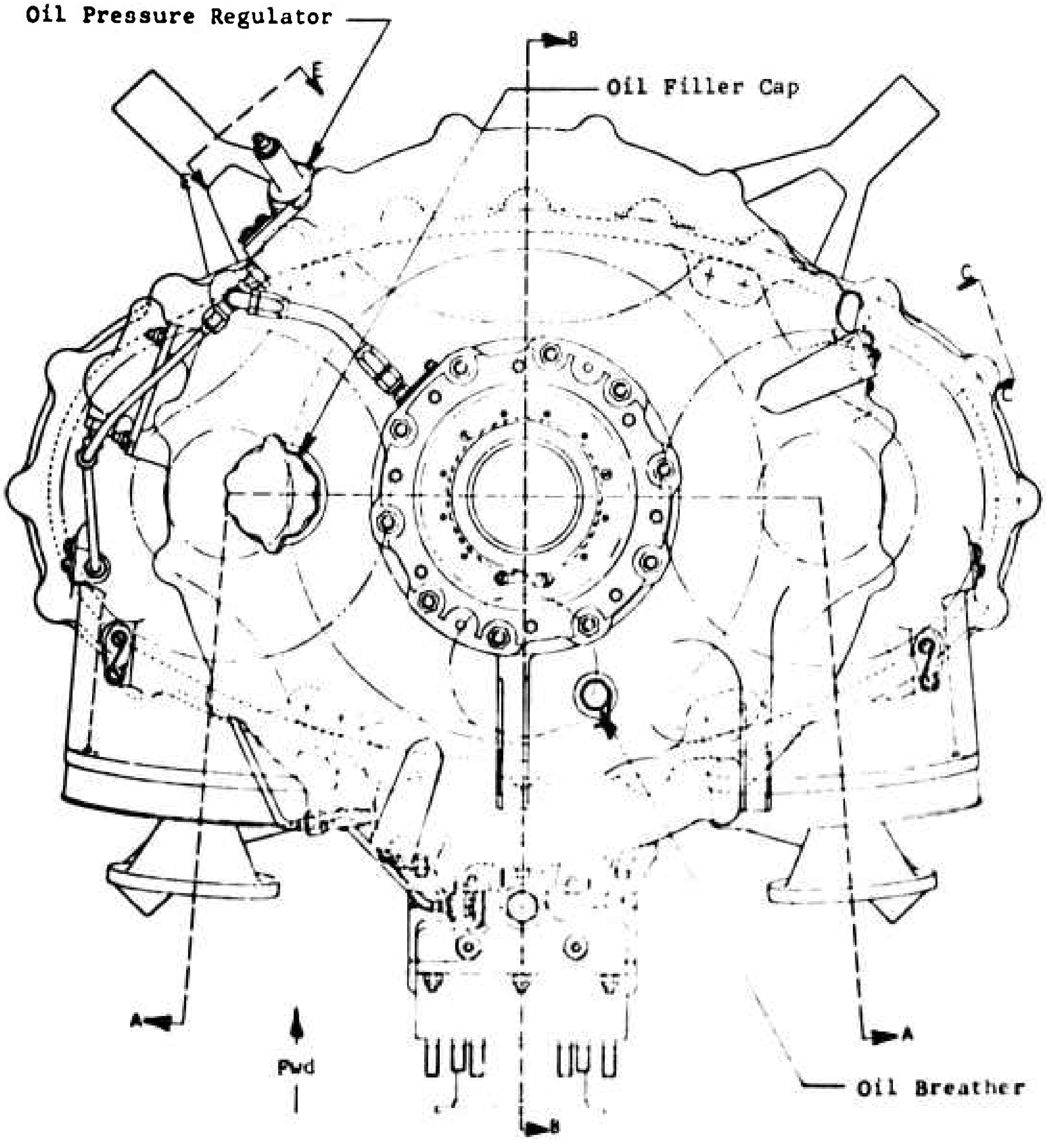

Figure 8. Twin 1500-HP Transmission, 5000-Hour TBO.

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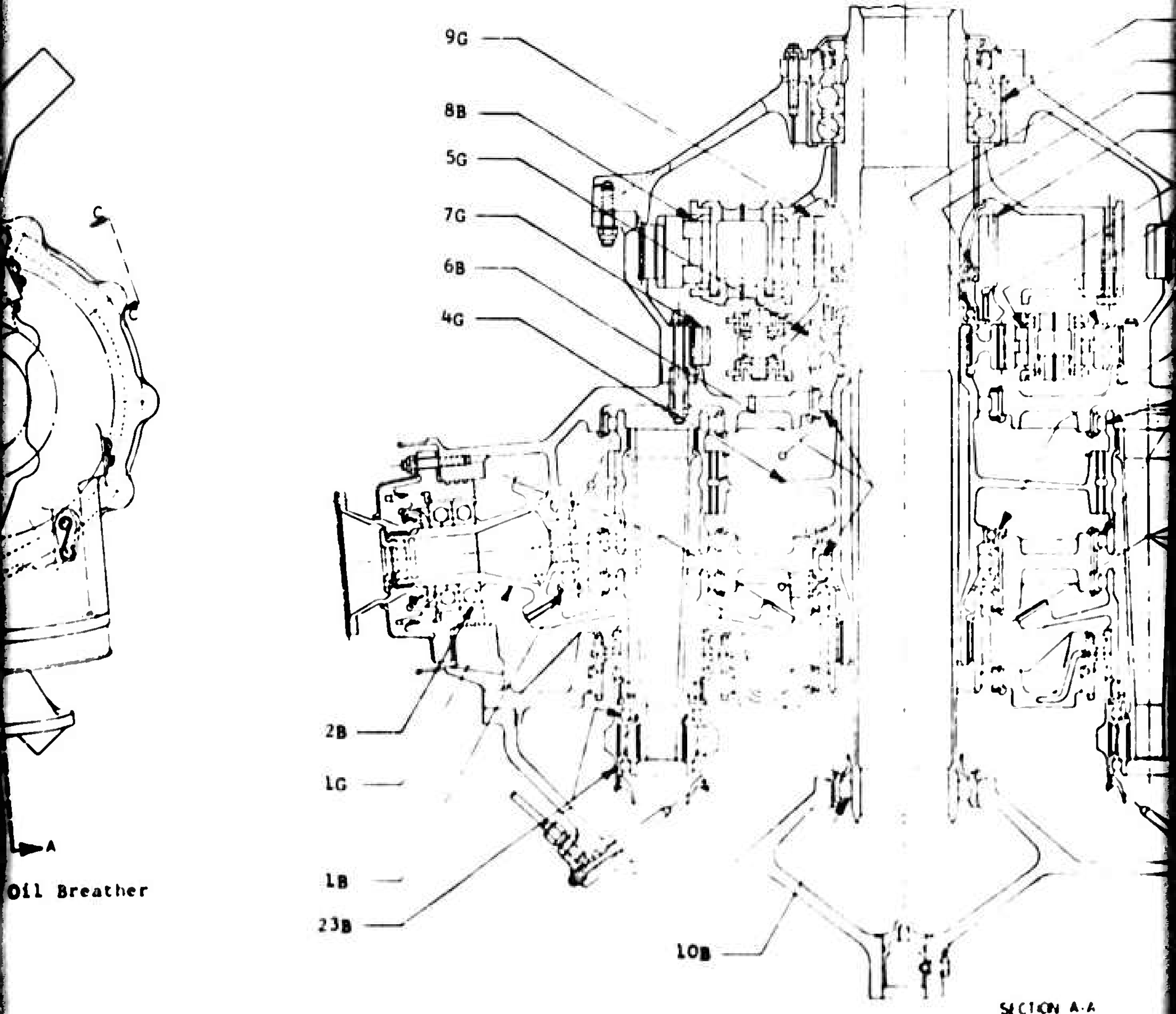

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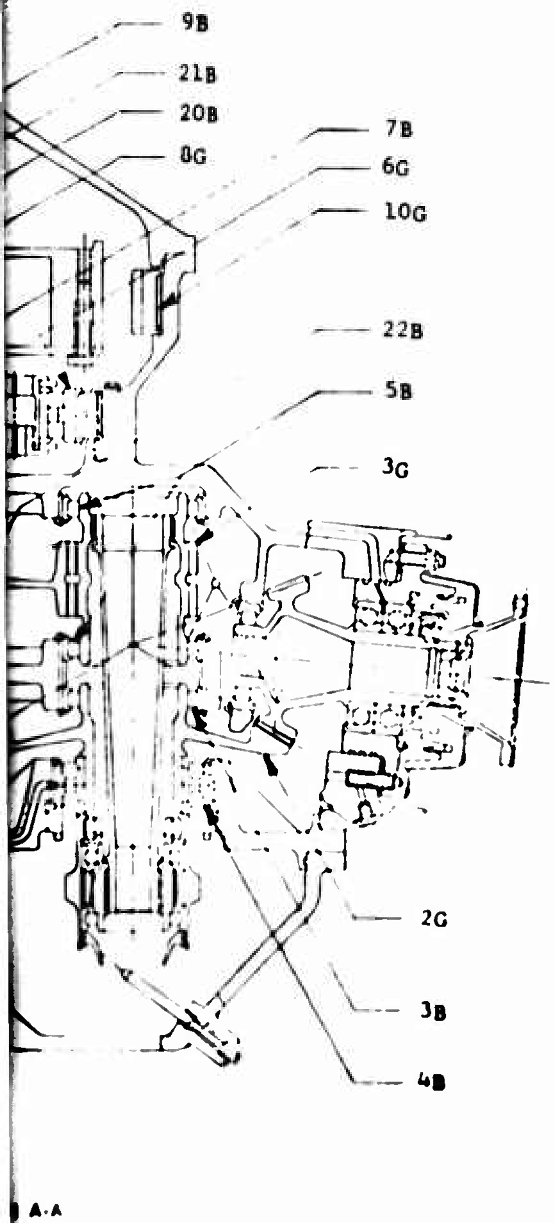

GEARS

Item No.	Name
1G	Input Spiral Bevel Pinion
2G	Input Spiral Bevel Gear
3G	Herringbone Pinion
4G	Herringbone Gear
5G	Lower Sun Gear
6G	Lower Planet Pinion
7G	Lower Ring Gear
8G	Upper Sun Gear
9G	Upper Planet Pinion
10G	Upper Ring Gear
11G	Offset Spur Pinion
12G	Offset Spur Gear
13G	Sump Bevel Pinion
14G	Sump Bevel Gear
15G	Accessory Spur Pinion
16G	Accessory Spur Idler
17G	Accessory Spur Gear

BEARINGS

Item No.	Name
1B	Input Pinicn Roller Size 207
2B	Input Pinion Ball Size 7210
3B	Input Gear Roller Size 1914
4B	Input Gear Ball Size 7014

D

RS

Name
Input Spiral Bevel Pinion
Input Spiral Bevel Gear
Herringbone Pinion
Herringbone Gear
Lower Sun Gear
Lower Planet Pinion
Lower Ring Gear
Upper Sun Gear
Upper Planet Pinion
Upper Ring Gear
Offset Spur Pinion
Offset Spur Gear
Sump Bevel Pinion
Sump Bevel Gear
Accessory Spur Pinion
Accessory Spur Idler
Accessory Spur Gear

NGS

Name
Input Pinion Roller Size 207
Input Pinion Ball Size 7210
Input Gear Roller Size 1914
Input Gear Ball Size 7014

BEARINGS - Continued

Item No.	Name
5B	Herringbone Pinion Roller Size 1914
6B	Herringbone Gear Roller 204-040-271
7B	Lower Planet Roller 204-040-725/-132
8B	Upper Planet Roller 563-040-112/-113
9B	Mast Ball 205-040-165
10B	Mast Roller 563-040-270
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11
12B	35T Spur Pinion Roller Size 206; 13 - 9 x 11
13B	Pinion Roller Size 208
14B	Pinion Ball Size 7212
15B	Gear Roller Size 211
16B	Gear Ball Size 7210
17B	34T Acc. Drive Roller Size 1004
18B	39T Acc. Drive Idler Roller Size 1905
19B	34T Acc. Drive (Single Mesh) Roller Size 1906
20B	Upper Planetary Support Ball 563-040-120
21B	Lower Planetary Support Ball 563-040-120
22B	Herringbone Gear Support Ball Size 140 x 165
23B	Freewheeling Roller Size 1009

A

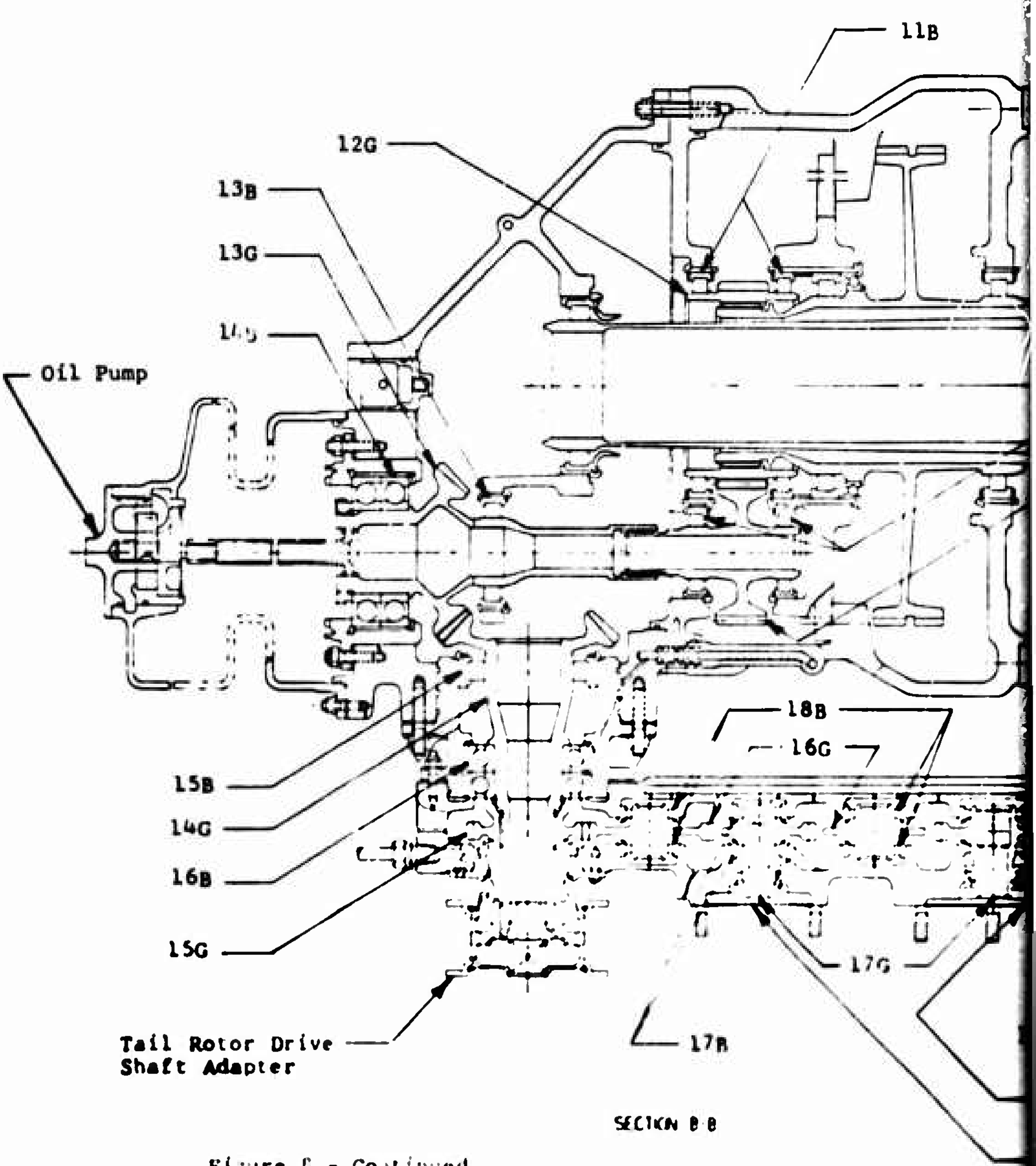

Figure 8 - Continued.

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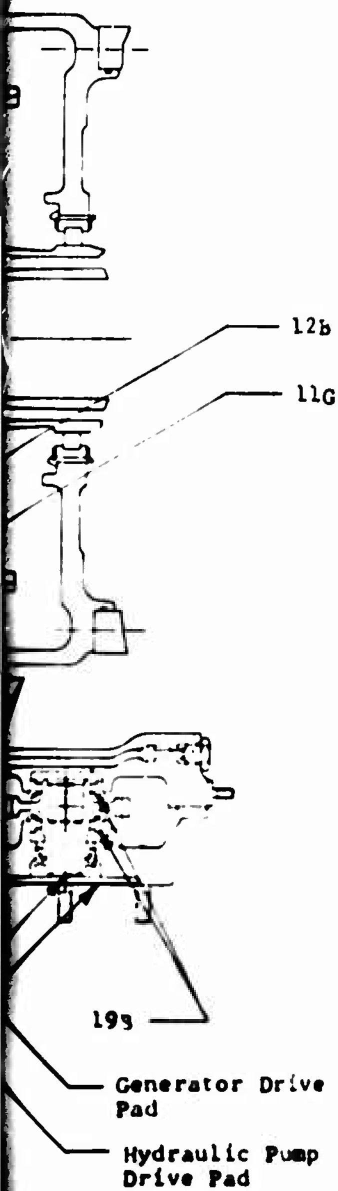
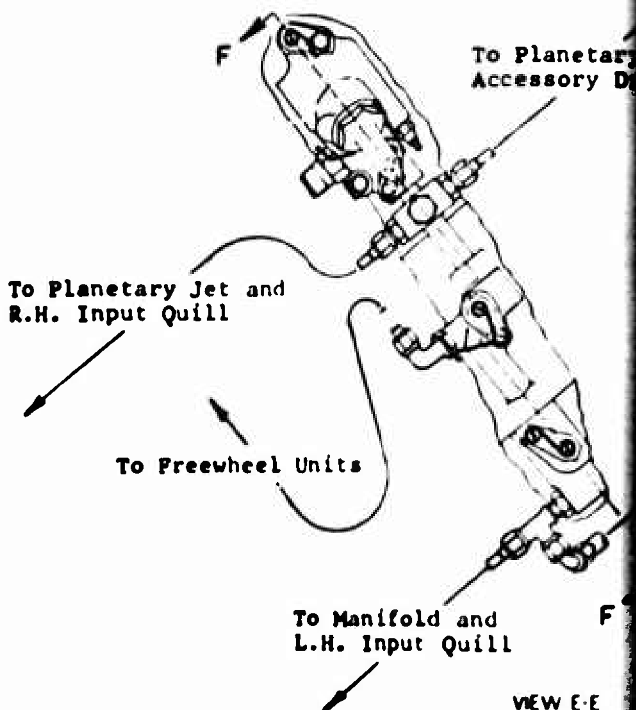
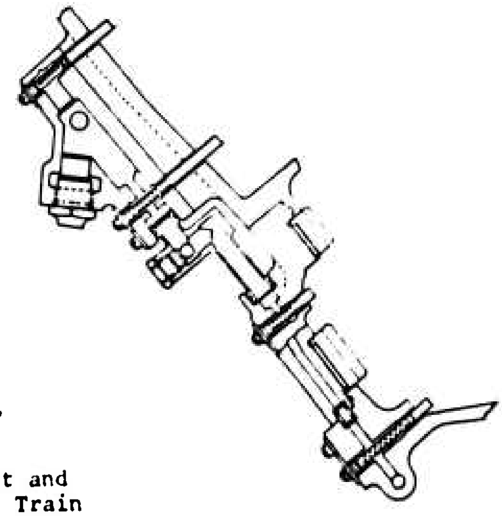

12b
11G
193
Generator Drive
Pad
Hydraulic Pump
Drive Pad

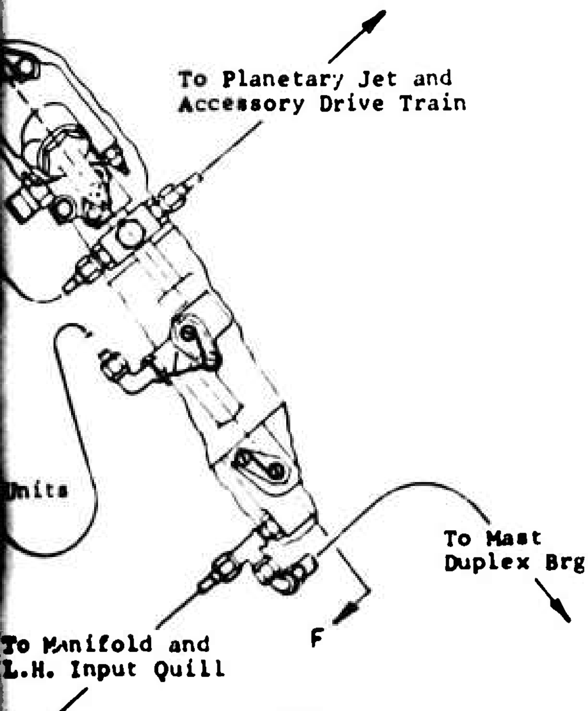

F
To Planetary
Accessory D
To Planetary Jet and
R.H. Input Quill
To Freewheel Units
To Manifold and
L.H. Input Quill
F

VIEW E-E

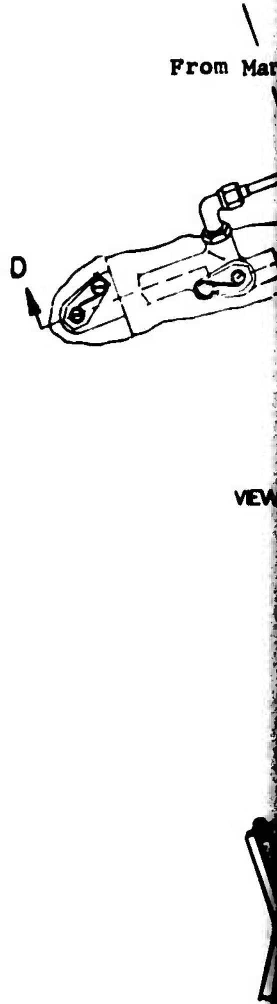
C



SECTION F-F

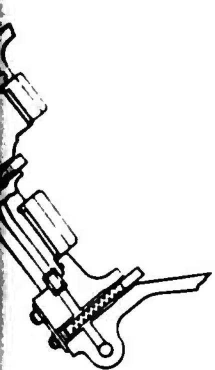


VIEW E-E

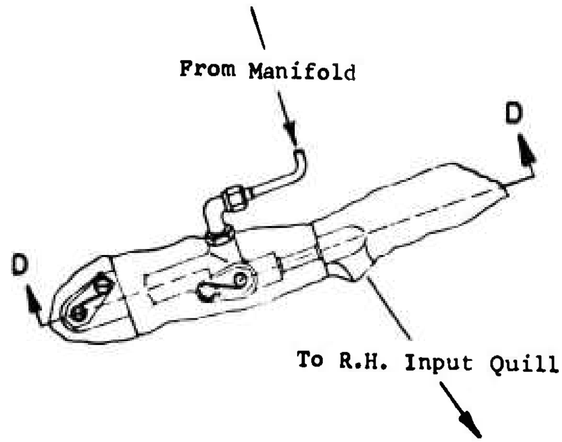


VIEW

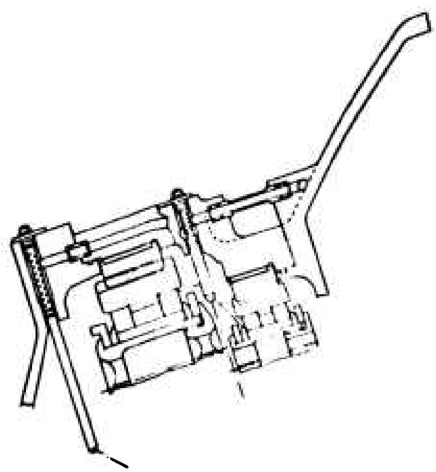
D



ECTION F-F



VIEW C-C



SECTION D-D

A

TABLE X. SUMMARY OF LOADS AND LIVES - 6000-HOU

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bearin Hou L1[illegible]
1B	Input Pinion Roller Size 207	-	916	-	9,
2B	Input Pinion Ball Size 7210	372	326	125	13,
3B	Input Gear Roller Size 1914	-	605	-	98,
4B	Input Gear Ball Size 7014	490	648	329	13,
5B	Herringbone Pinion Roller Size 1914	-	1,652	-	11,
6B	[illegible]ngbone Gear Roller 204 [illegible]271	-	1,652	-	32,
7B	Lower Pl[illegible]t Roller 204-040-72[illegible]2	-	2,025	-	18,
8B	Upper Planet Rol[illegible]r 563-040-112/-113	-	4,660	-	10,
9B	Mast Ball 205-040-165	5,176	2,036	3,272	10,
10B	Mast Roller 563-040-270	-	2,668	-	46,
11B	51T Spur Gear Roller 165 x 16; 33 - 11 x 11	-	1,125	-	71,
12B	35T Spur Pinion Roller Size 206; 13 - 9 x 11	-	1,125	-	11,
13B	Pinion Ball Size 208	-	1,52[illegible]	-	23,

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13

ADS AND LIVES - 6000-HOUR TBO, TWIN 1500 HP

ing * al)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
16	-	9,080	24,000	43	840,000
26	125	13,376	24,000	58	1,2[illegible],000
05	-	98,304	11,871	37	830,970
48	329	13,244	11,871	41	830,970
52	-	11,920	11,871	35	830.970
52	-	32,964	3,011	22	361,320
25	-	18,368	3,743	11	131,005
60	-	10,736	808	5	44,440
36	3,272	10,588	252	14	27,720
68	-	46,037	252	3	24,192
25	-	71,591	3,011	7	361,320
25	-	11,896	4,387	15	131,610
20	-	23,736	4,387	6	175,480

A

TABLE X - Con

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bea
14B	Pinion Ball Size 7212	1,643	827	800	1
15B	Gear Roller Size 211	-	3,407	-	1
16B	Gear Ball Size 7210	504	670	285	1
17B	34T Accessory Drive Roller Size 1004	-	498	-	1
18B	39T Accessory Drive Idler Roller Size 1905	-	498	-	1
19B	34T Accessory Drive (Single Mesh) Roller Size 1906	-	314	-	1
20B	Upper Planetary Support Ball 563-040-120	-	-	-	3,2
21B	Lower Planetary Support Ball 563-040-120	-	-	-	5
22B	Herringbone Gear Support Ball Size 140 x 165	-	-	-	9
23B	Freewheeling Roller Size 1009	-	-	-	

* 65% Items 1 through 10
75% Items 11 through 19
100% Items 20 through 23

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B

TABLE X - Continued

ing * ial b)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
827	800	13,112	4,387	8	263,220
407	-	13,592	4,150	23	228,250
670	285	17,344	4,150	7	207,500
498	-	19,288	4,150	20	83,000
498	-	11,664	3,618	10	90,450
314	-	17,562	4,150	22	124,500
-	-	3,280,000	723	13	57,840
-	-	512,260	2,036	15	210,770
-	-	91,645	3,011	15	210,770
-	-	-	-	-	-

A

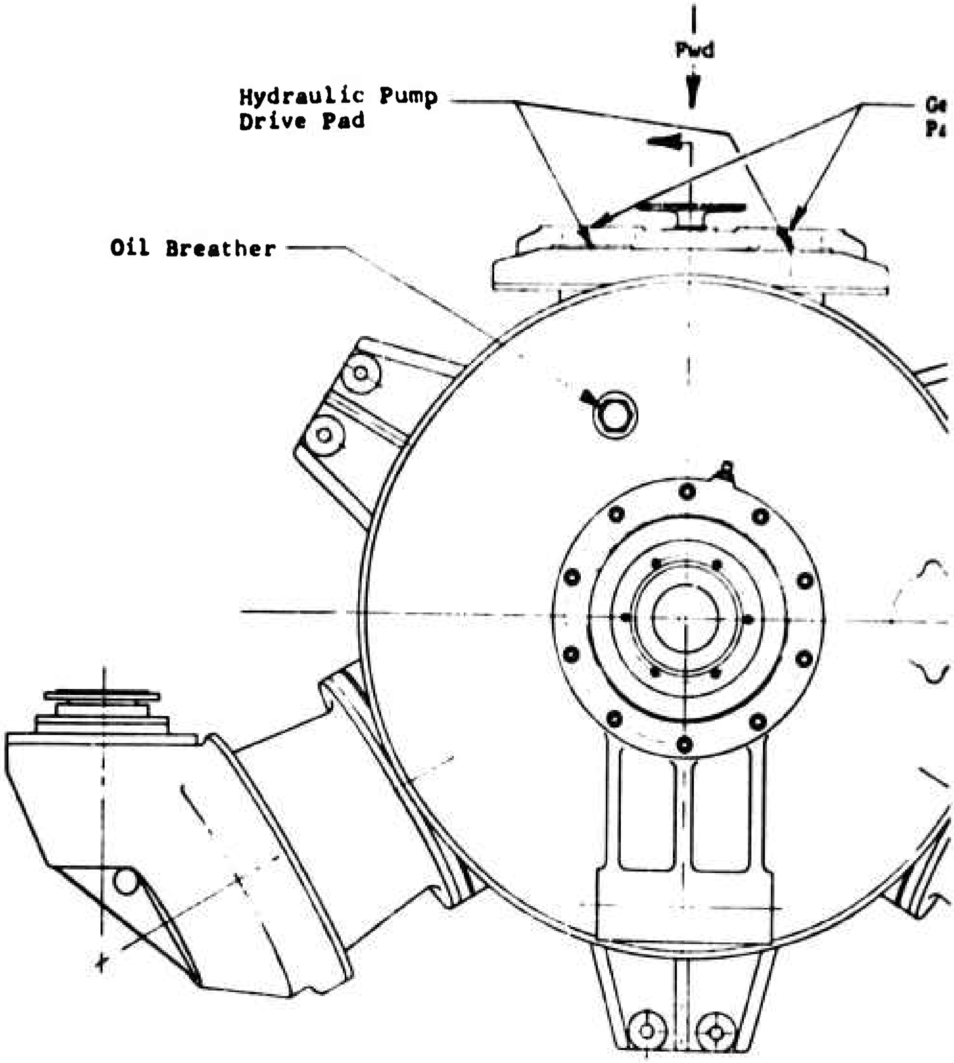
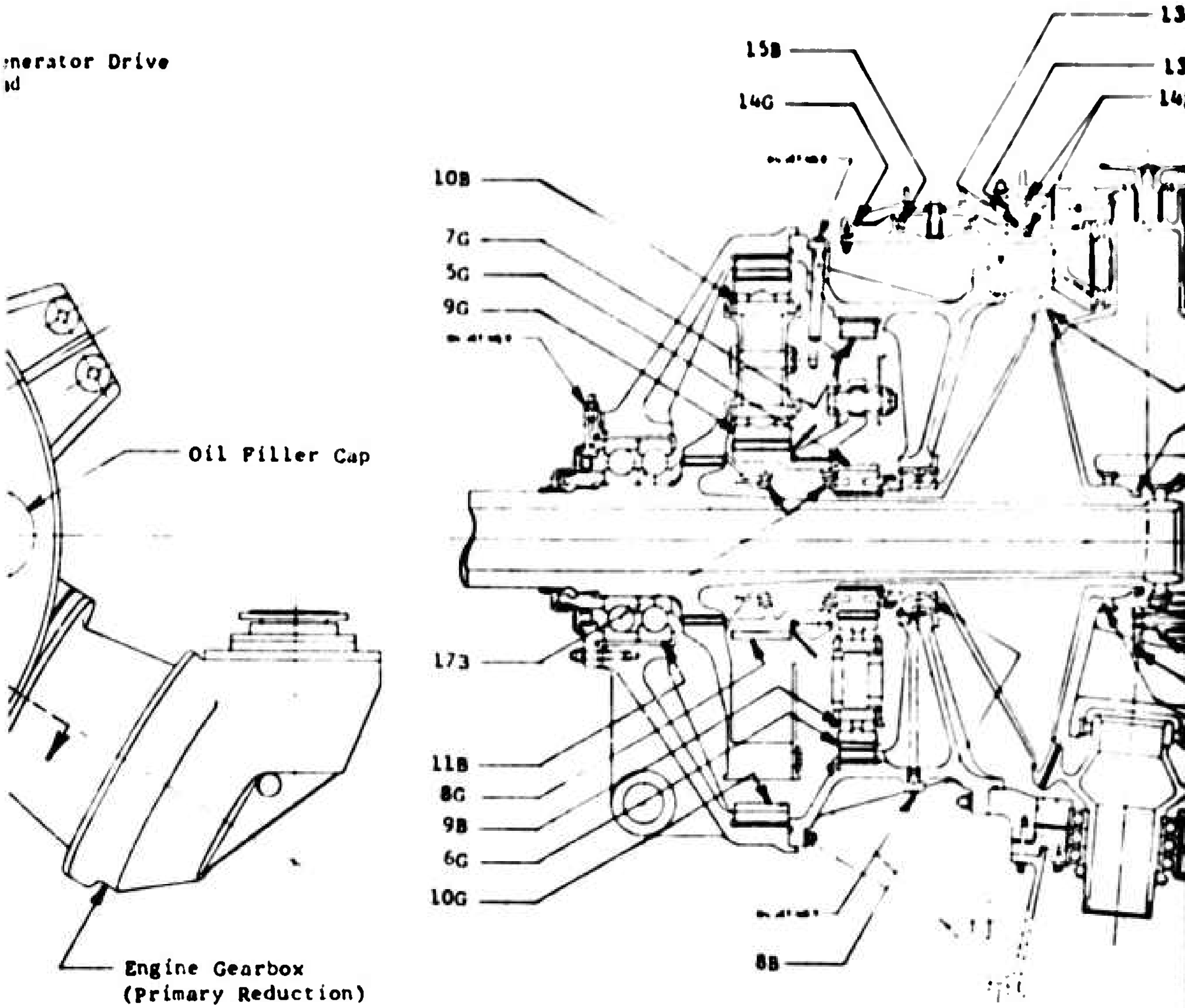
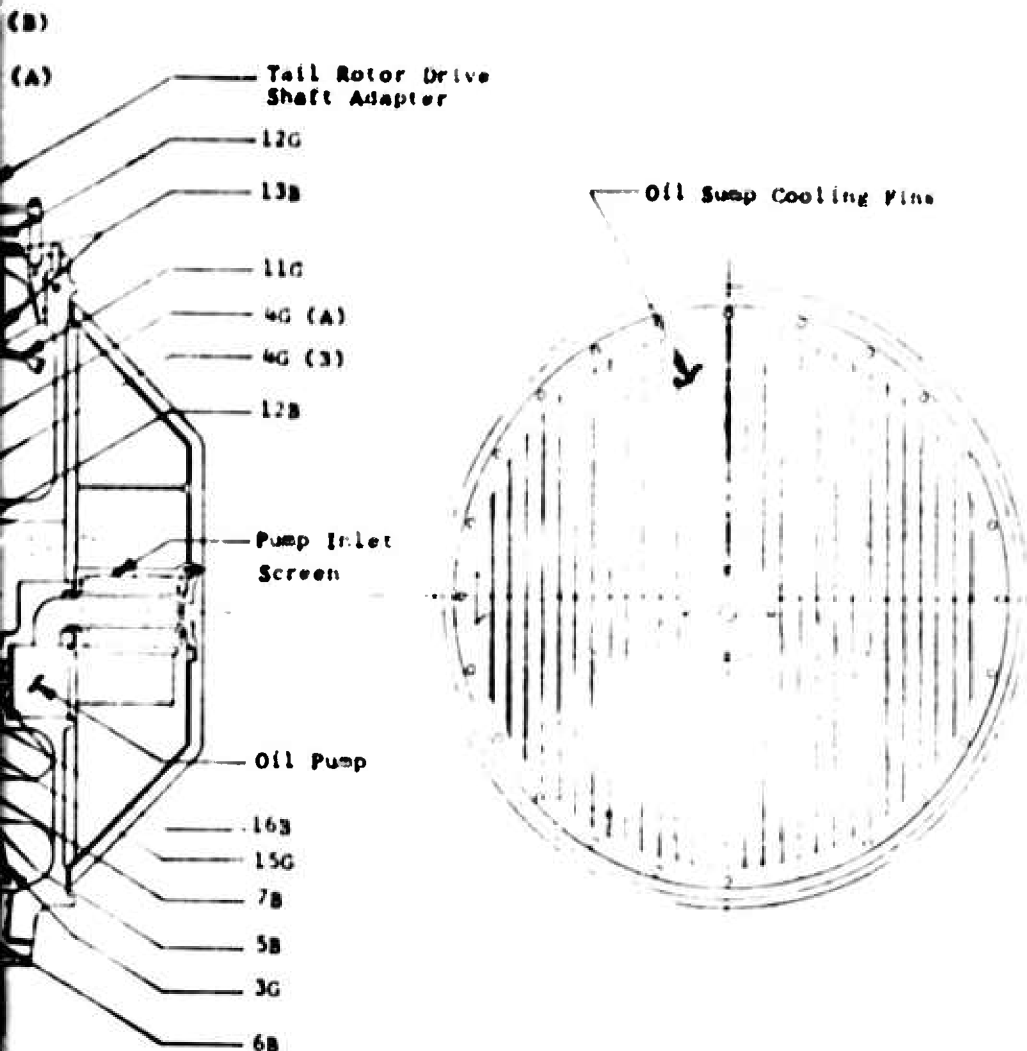


Figure 9. Twin 4800-HP Transmission, 1200-Hour T30.

B



nerator Drive
15B
14G
10B
7G
5G
9G
Oil Filler Cap
17B
11B
8G
9B
6G
10G
8B
Engine Gearbox
(Primary Reduction)



GEARS

Item No.		Name
1G		Engine Grbx
2G		Engine Grbx
3G		Input Spiral
4G	(A)	Input Spiral
	(B)	Oil Pump Ta
5G		Lower Sun G
6G		Lower Plane
7G		Lower Ring
8G		Upper Sun G
9G		Upper Plane
10G		Upper Ring
11G		Tail Rotor Takeoff Sp
12G		Gen. and Hy Takeoff Sp
13G	(A)	Generator
	(B)	Hydr. Pump
14G		Hydr. Pump
15G		Oil Pump D

BEARINGS

Item No.	Name
1B	Engine Gea Roller Si
2B	Engine Gea Ball Size
3B	Engine Gea Roller Si

GEARS

No.	Name
	Engine Grbx. Spiral Bevel Pinion
	Engine Grbx. Spiral Bevel Gear
	Input Spiral Bevel Pinion
(A)	Input Spiral Bevel Gear
(B)	Oil Pump Takeoff Spur
	Lower Sun Gear
	Lower Planet Pinion
	Lower Ring Gear
	Upper Sun Gear
	Upper Planet Pinion
	Upper Ring Gear
	Tail Rotor and Access Drive Takeoff Spiral Bevel
	Gen. and Hydr. Pump Drive Takeoff Spur
(A)	Generator Drive Spur
(B)	Hydr. Pump Drive Takeoff Spur
	Hydr. Pump Drive Spur
	Oil Pump Drive Spur

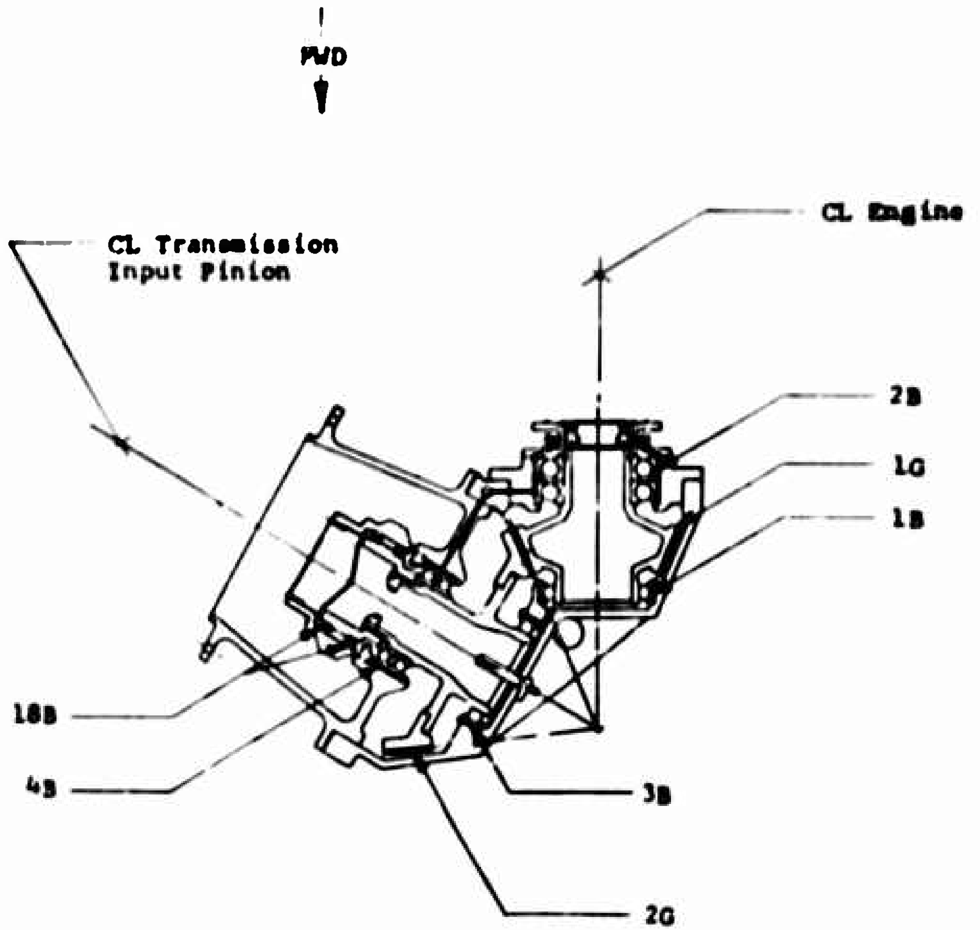
BEARINGS

No.	Name
	Engine Gearbox Pinion Roller Size 1020
	Engine Gearbox Pinion Ball Size 7218
	Engine Gearbox Gear Roller Size 1020

BEARINGS - Continued

Item No.	Name
4B	Engine Gearbox Gear Ball Size 7018
5B	Input Pinion Roller Size 1924
6B	Input Pinion Ball Size 7024
7B	Input Gear Roller Size 36 - 12.5 x 14 x 7.3425 PD
8B	Input Gear Ball Size 1932
9B	Lower Planet Roller Size 17 x 19 x 5.5707 PD 2 Rows: 19 Rollers/Row
10B	Upper Planet Roller Size 1/2 x .85 x 7.500 PD 2 Rows: 30 Rollers/Row
11B	Mast Ball Size 7234-9234
12B	Mast Roller Size 24 - 15 x 17 x 5.4925 PD
13B	Tail Rotor and Access Bevel Ball Size 7124
14B	Generator Drive Ball Size KD40CP (Kaydon)
15B	Hydr. Pump Drive Ball Size KD40CP (Kaydon)
16B	Oil Pump Drive Ball Size 26 x 52 x 15
17B	Planetary Support Ball Size AF65AH (Kaydon)
18B	Freewheeling Ball Size KD45XP (Kaydon)

1



SECTION THRU INPUT REDUCTION GEARBOX

Figure 9 - Continued.

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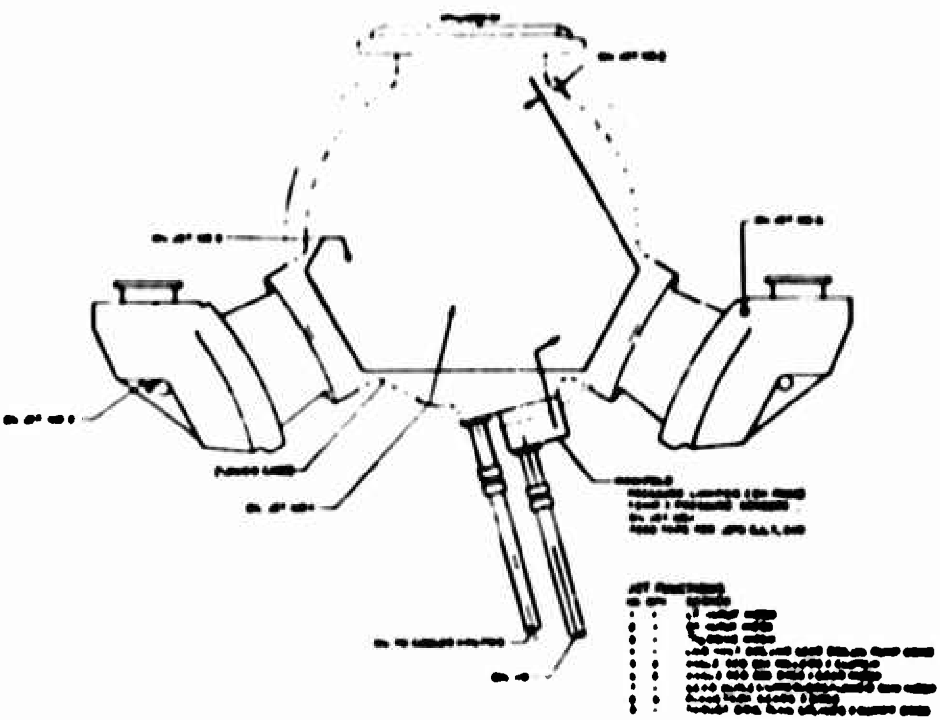
13

CL Engine

2B

1G

1B



LUBRICATION SCHEMATIC

A

TABLE XI. SUMMARY OF BEARING LOADS AND LI

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)
1B	Engine Gearbox Pinion Roller Size 1020	-	3,495	-
2B	Engine Gearbox Pinion Ball Size 7128	1,717	2,029	1,543
3B	Engine Gearbox Gear Roller Size 1020	-	4,070	-
4B	Engine Gearbox Gear Ball Size 7018	724	1,367	862
5B	Input Pinion Roller Size 1020	-	3,275	-
6B	Input Pinion Ball Size 7024	2,036	1,487	2,497
7B	Input Gear Roller Size 36-12.5 x 14 x 7.3425 PD	-	4,992	-
8B	Input Gear Ball Size 1932	2,762	2,332	4,162
9B	Lower Planet Roller 2 Rows: 19 Rollers/Row Size 17 x 19 x 5.5707 PD	-	6,461	-
10B	Upper Planet Roller 2 Rows: 30 Rollers/Row Size 1/2 x .85 x 7.500 PD	-	10,610	-
11B	Mast Ball Size 7234-9234	25,065	9,385	27,296
12B	Mast Roller Size 24-15 x 17 x 5.4925 PD	-	12,306	-

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B

RING LOADS AND LIVES - 1200-HOUR TBO, TWIN 4800 HP

ng • al b	Moment (in.-lb)	Bearing Life Hours L10 •	RPM	Minimum Film Thickness (10^{-6} in.)	DN
95	-	4,916	9,000	43.6	900,000
29	1,543	2,924	9,000	48.99	810,000
70	-	2,750	6,090	32.6	609,000
67	862	3,088	6,090	32.2	548,100
75	-	3,686	6,090	35.7	730,800
87	2,497	2,774	6,090	42.2	730,800
92	-	3,086	2,268	20.9	365,960
32	4,162	3,336	2,268	24.7	362,880
61	-	3,240	1,293	11.7	138,853
10	-	2,842	334	5.1	55,102
35	27,296	4,716	151	5.4	25,670
06	-	2,822	151	2.3	16,520

TABLE XI - Conti[nued]

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)
13B	Tail Rotor and Access Bevel Ball Size 7124	721	2,202	1,714
14B	Generator Drive Ball Size KD40CP, 2 on Shaft (Kaydon)	- -	71 156	4 4
15B	Hydraulic Pump Drive Ball Size KD40CP (Kaydon)	-	53	-
16B	Oil Pump Drive Ball Size 26 x 52 x 15	-	-	-
17B	Planetary Support Ball Size KF65AH (Kaydon)	-	-	-
18B	Freewheeling Ball Size KD45XP (Kaydon)	-	-	-

* 65% - Items 1B through 10B
100% - Items 11B and 12B
75% - Items 13B through 15B
No calculations are made for items 16B through 18B

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B

TABLE XI - Continued

ng * al)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
02	1,714	2,808	4,943	34.9	593,160
71	4	7,520 (1)	6,355	28.5	645,178
56	4	6,060 (2)		28.1	
33	-	12,732	3,813	19.7	387,107
	-	-	-	-	-
	-	-	-	.	-
	-	-	-	-	-

A

TABLE XII. SUMMARY OF
TWIN 4800 H

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Velo (ft/
1G	Engine Gear-box Spiral Bevel Pinion	4,800	8,797	7.75	67	2.500	9,000	20,3
2G	Engine Gear-box Spiral Bevel Gear	4,800	8,797	7.75	99	2.500	6,091	20,3
3G	Input Spiral Bevel Pinion	4,800	9,770	7.75	86	2.750	6,091	17,48
4G(A)	Input Spiral Bevel Gear	4,800	9,770	7.75	231	2.750	2,268	17,48
(B)	Oil Pump Takeoff Spur	10	93	10.00	60	.500	2,268	3,5
5G	Lower Sun Gear	6,400	9,970	5.3817	48	2.600	2,268	-
6G	Lower Planet Pinion							
	With Sun Gear	6,400	9,970	5.3817	43	2.400	1,878	4,0
	With Ring Gear	6,400	9,970	5.6181	43	2.400	1,878	4,0
7G	Lower Ring Gear	6,400	9,970	5.6181	138	2.200	-	-
8G	Upper Sun Gear	6,400	19,474	4.1586	48	3.800	585	-
9G	Upper Planet Pinion							
	With Sun Gear	6,400	19,474	4.1586	43	3.650	485	1,31
	With Ring Gear	6,400	19,474	4.3414	43	3.650	485	1,31

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B

ARY OF FUNCTIONAL GEAR DATA,
4800 HP, 1200-HOUR TBO

Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
20,370	132	38,971	185,595	750	13.335	2
20,370	132	36,714	189,595	750	13.335	2
17,486	114	38,571	147,022	750	8.402	-
17,486	114	37,861	147,022	750	8.402	2
3,563	43	4,522	73,720	29.05	.032	1
-	68	38,617	163,487	28.5	1.274	4
4,055	68	48,541	163,487	28.5	1.274	-
4,055	31	36,418	146,457	29.5	1.298	-
-	31	53,381	146,457	29.5	1.298	4
-	67	39,569	162,874	16.5	1.269	6
1,319	67	47,207	162,874	16.5	1.269	-
1,319	32	38,535	142,656	17.1	1.358	-

A

TABLE XII - C

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Veloc (ft/m
10G	Upper Ring Gear	6,400	19,474	4.3414	138	3.500	--	--
11G	Tail Rotor & Acc. Drive Takeoff Spiral Bevel	1,010	2,206	8.856	106	.700	4,943	15,49
12G	Gen. & Hydr. Pump Drive Takeoff Spur	65	184	8.0	72	.400	4,943	11,64
13G(A)	Gen. Drive Spur	65	184	8.0	56	.300	6,355	11,64
(B)	Hydr. Pump Drive Takeoff Spur	15	50	8.0	48	.300	6,355	9,98
14G	Hydr. Pump Drive Spur	15	50	8.0	80	.20	3,813	9,98
15G	Oil Pump Drive Spur	10	93	10.0	36	.25	3,780	3,50

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B

BLE XII - Continued

RPM	Pitch Line Velocity (ft/min)	Temperature Rise (^oF)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss (hp)	Number of Meshes
--	--	32	50,575	142,656	17.1	1.358	6
4,943	15,490	97	42,462	141,536	750	2.112	1
4,943	11,647	36	7,567	73,347	750	.074	2
6,355	11,647	36	10,021	73,347	750	.074	2
6,355	9,982	16	2,626	45,393	50.1	.020	2
3,813	9,982	16	3,734	46,393	50.1	.020	2
3,780	3,563	43	8,038	73,720	29.05	.032	1

A

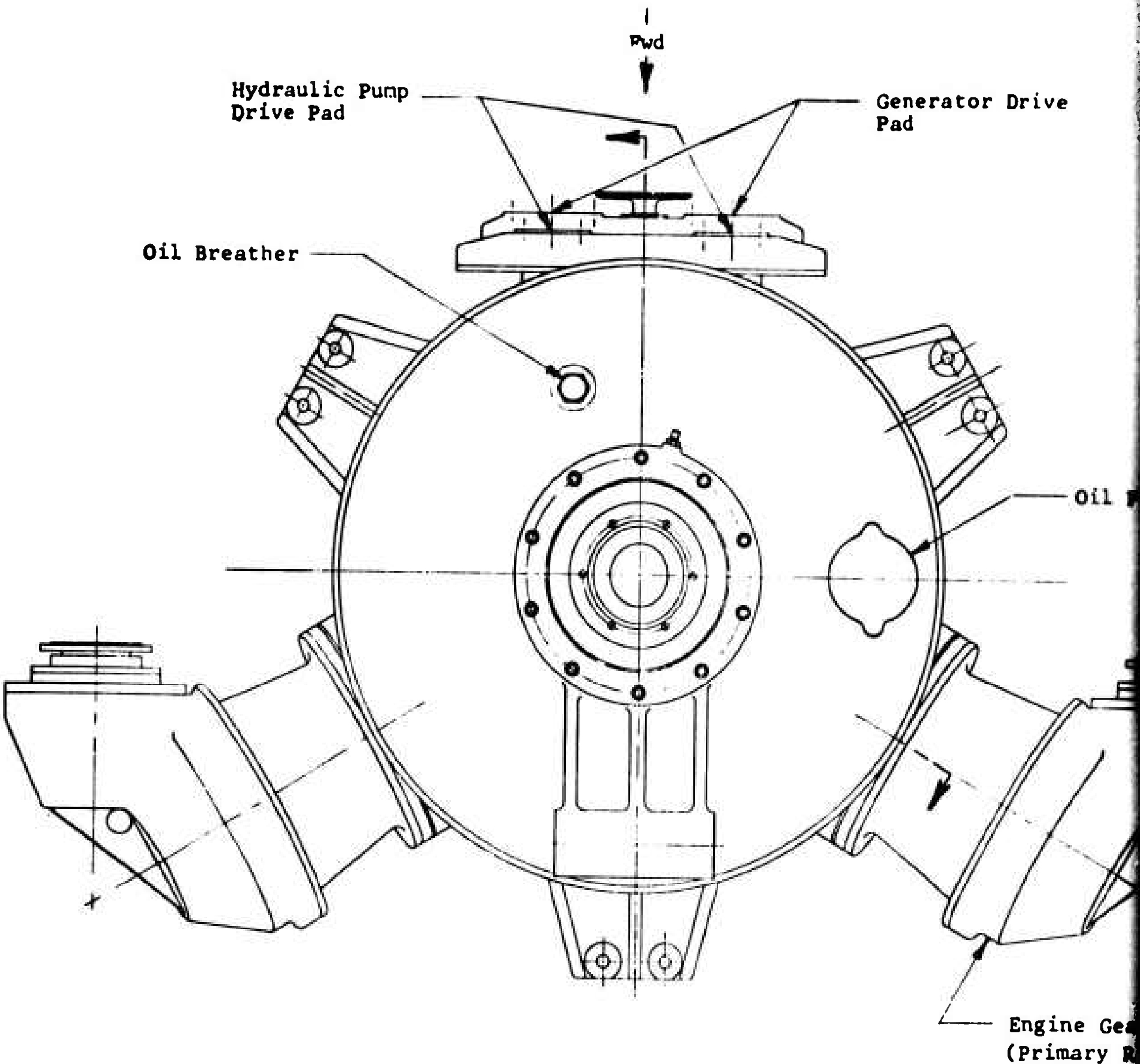

Figure 10. Twin 4000-HP Transmission, 3000-Hour TBO.

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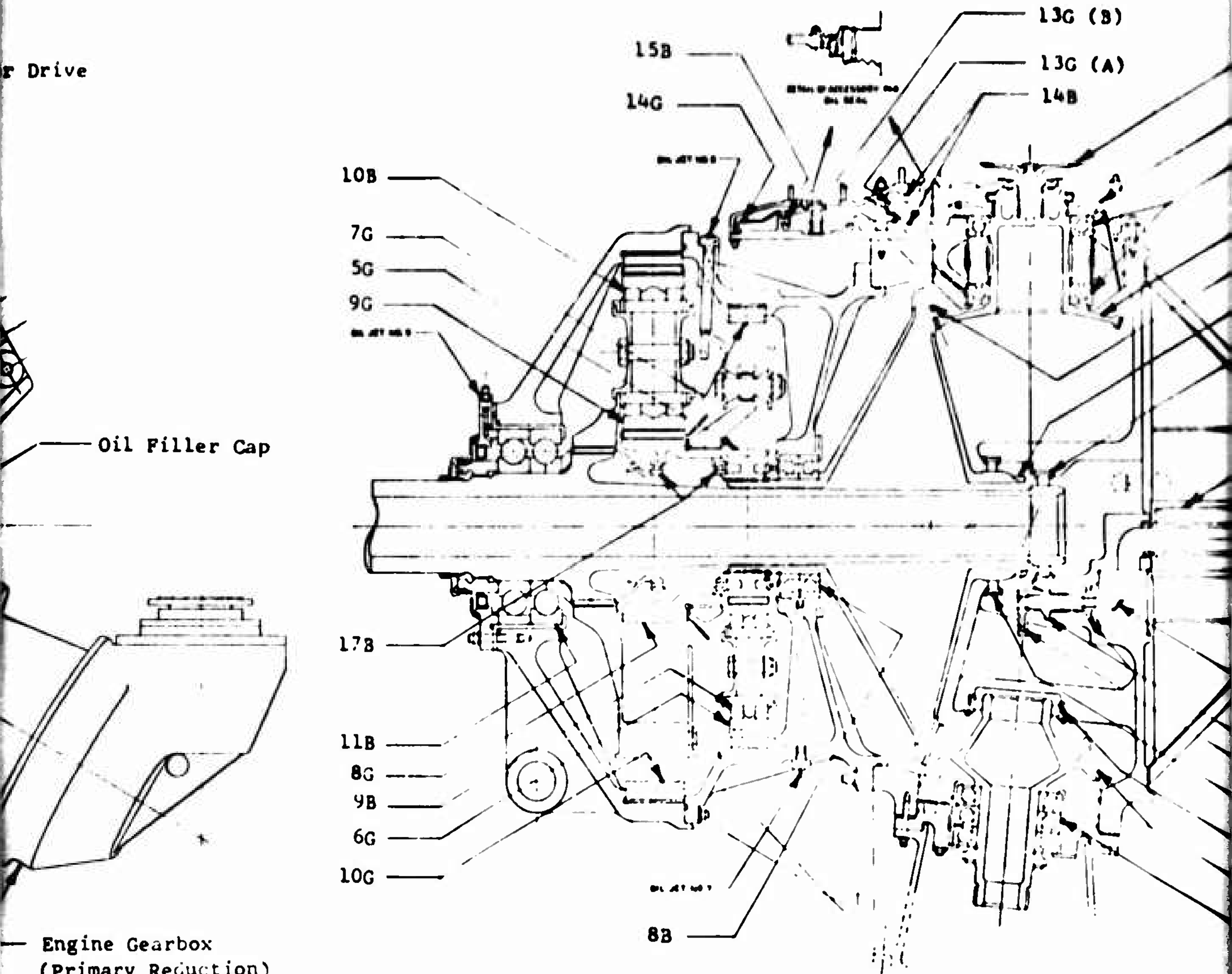
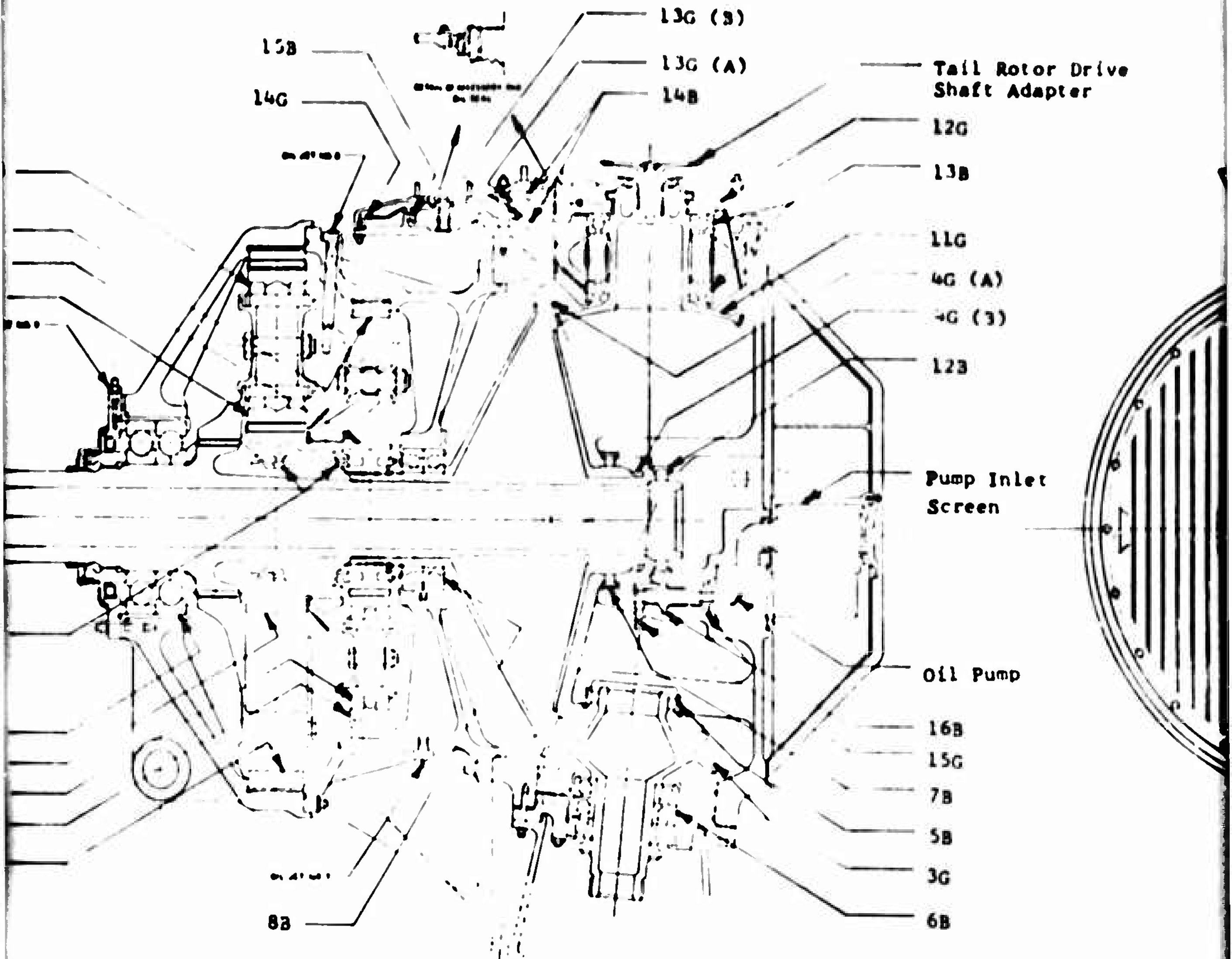
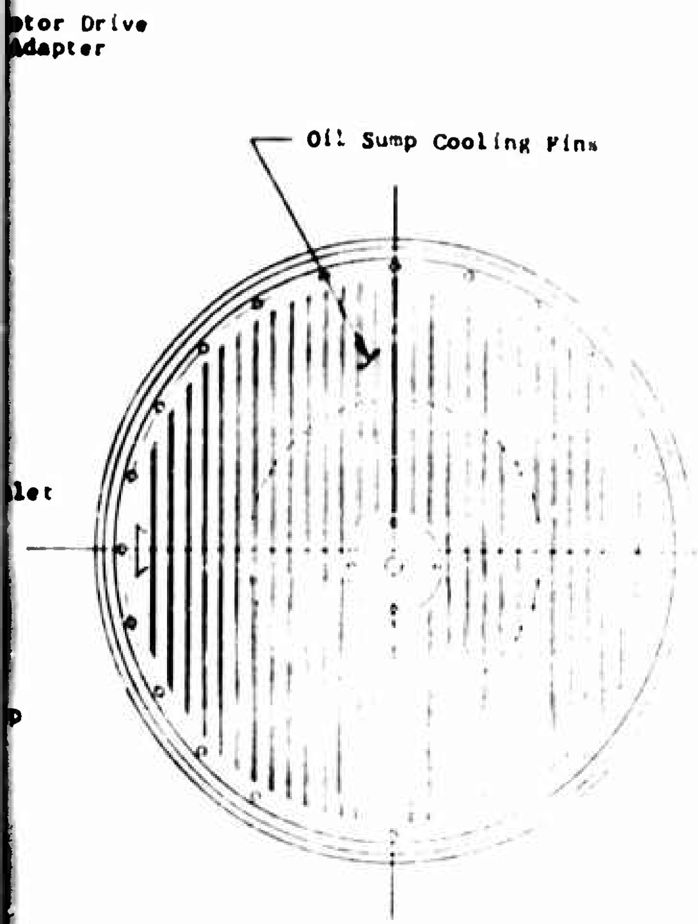


r Drive
Oil Filler Cap
Engine Gearbox
(Primary Reduction)
13G (B)
13G (A)
14B
15B
14G
10B
7G
5G
9G
17B
11B
8G
9B
6G
10G
8B


13G (B)
15B
13G (A)
Tail Rotor Drive Shaft Adapter
14G
14B
12G
13B
11G
4G (A)
4G (3)
12B
Pump Inlet Screen
Oil Pump
16B
15G
7B
5B
3G
6B
8B



GEARS

Item No.		Name
1G		Engine Grbx. Spiral Bevel Pinion
2G		Engine Grbx. Spiral Bevel Gear
3G		Input Spiral Bevel Pinion
4G	(A)	Input Spiral Bevel Gear
	(B)	Oil Pump Takeoff Spur
5G		Lower Sun Gear
6G		Lower Planet Pinion
7G		Lower Ring Gear
8G		Upper Sun Gear
9G		Upper Planet Pinion
10G		Upper Ring Gear
11G		Tail Rotor and Access Drive Takeoff Spiral Bevel
12G		Gen. and Hydr. Pump Drive Takeoff Spur
13G	(A)	Generator Drive Spur
	(B)	Hydr. Pump Drive Takeoff Spur
14G		Hydr. Pump Drive Spur
15G		Oil Pump Drive Spur

BEARINGS

Item No.	Name
1B	Engine Grbx. Pinion Roller Size 1020
2B	Engine Grbx. Pinion Ball Size 7220
3B	Engine Grbx. Gear Roller Size 1021

GEARS

Item No.		Name
1G		Engine Grbx. Spiral Bevel Pinion
2G		Engine Grbx. Spiral Bevel Gear
3G		Input Spiral Bevel Pinion
4G	(A)	Input Spiral Bevel Gear
	(B)	Oil Pump Takeoff Spur
5G		Lower Sun Gear
6G		Lower Planet Pinion
7G		Lower Ring Gear
8G		Upper Sun Gear
9G		Upper Planet Pinion
10G		Upper Ring Gear
11G		Tail Rotor and Access Drive Takeoff Spiral Bevel
12G		Gen. and Hydr. Pump Drive Takeoff Spur
13G	(A)	Generator Drive Spur
	(B)	Hydr. Pump Drive Takeoff Spur
14G		Hydr. Pump Drive Spur
15G		Oil Pump Drive Spur

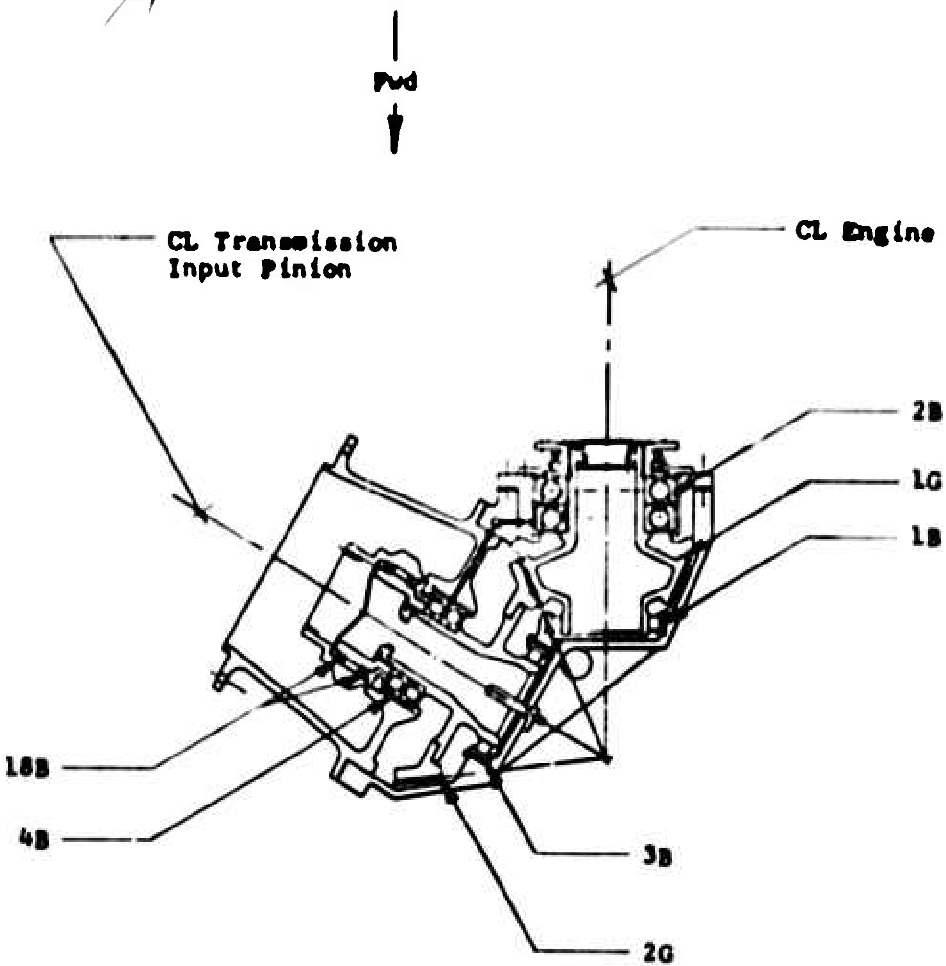
BEARINGS

Item No.	Name
1B	Engine Grbx. Pinion Roller Size 1020
2B	Engine Grbx. Pinion Ball Size 7220
3B	Engine Grbx. Gear Roller Size 1021

BEARINGS - Continued

Item No.	Name
4B	Engine Grbx. Gear Ball Size 7216
5B	Input Pinion Roller Size 1020
6B	Input Pinion Ball Size 7219
7B	Input Gear Roller Size 30 - 15 x 15 x 7.44 PD
8B	Input Gear Ball Size 7030
9B	Lower Planet Roller Size 21 x 21 x 5.6928 PD 2 Rows: 16 Rollers/Row
10B	Upper Planet Roller Size 19 x 26 x 7.252 PD 2 Rows: 24 Rollers/Row
11B	Mast Ball Size 7234-9234
12B	Mast Roller Size 1022
13B	Tail Rotor and Access Bevel Ball Size 7026
14B	Generator Drive Ball Size KD40CP (Kaydon)
15B	Hydr. Pump Drive Ball Size KD40CP (Kaydon)
16B	Oil Pump Drive Ball Size 26 x 52 x 15
17B	Planetary Support Ball Size KF65AH (Kaydon)
18B	Freewheeling Ball Size KD45XP (Kaydon)

A



SECTION THRU INPUT REDUCTION GEARBOX

Figure 10 - Continued.

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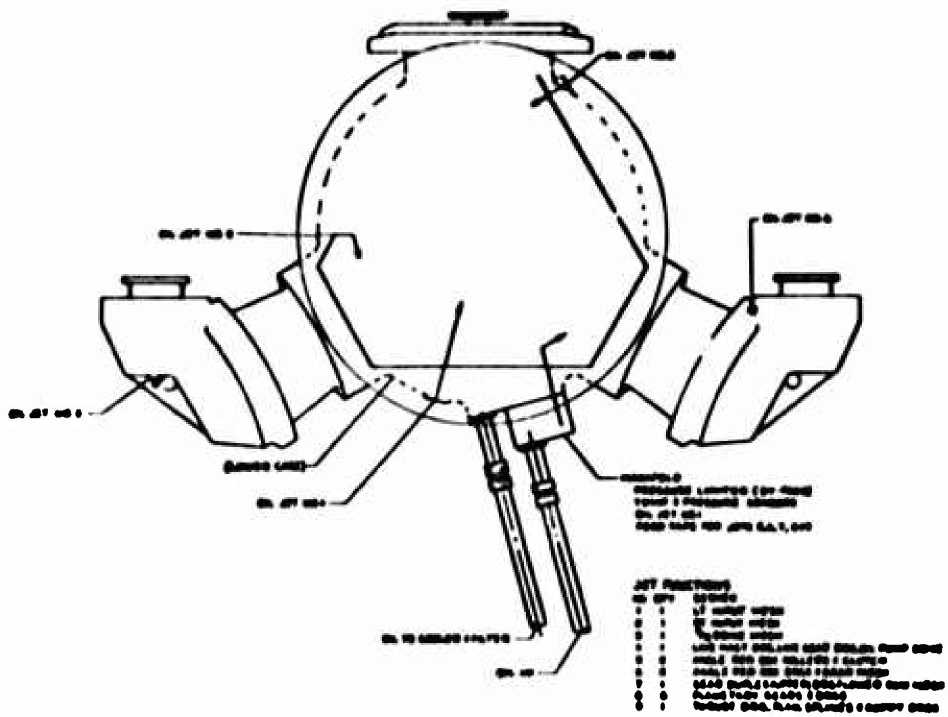
CL Engine

2B

1G

1B



LUBRICATION SCHEMATIC

/1

TABLE XIII. SUMMARY OF BEARING LOADS AND LIVES

			Loading *		Bear
Item	Bearing	Axial	Radial (lb)	Moment (in.-lb)	Ho Li
1B	Engine Gearbox Pinion Roller Size 1020	-	3,514	-	5,
2B	Engine Gearbox Pinion Ball Size 7220	1,489	1,986	1,600	6,
3B	Engine Gearbox Gear Roller Size 1021	-	4,014	-	5,
4B	Engine Gearbox Gear Ball Size 7216	712	1,425	876	5,
5B	Input Pinion Roller Size 1020	-	3,423	-	5,
6B	Input Pinion Ball Size 7219	3,052	1,446	1,714	5,
7B	Input Gear Roller Size 30-15 x 15 x 7.44 PD	-	5,001	-	5,
8B	Input Gear Ball Size 7030	2,747	2,390	4,223	5,
9B	Lower Planet Roller 2 Rows: 16 Rollers/Row Size 21 x 21 x 5.6928 PD	-	6,999	-	6,
10B	Upper Planet Roller 2 Rows: 24 Rollers/Row Size 19 x 26 x 7.252 PD	-	14,189	-	5,
11B	Mast Ball Size 7234-9234	25,065	9,901	27,712	5,
12B	Mast Roller Size 1022	-	12,392	-	5,

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13

ARING LOADS AND LIVES - 3000-HOUR TBO, TWIN 4800 HP

ing * ial lb)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
,514	-	5,472	9,000	55.8	900,000
,986	1,600	6,492	9,000	55.8	900,000
,014	-	5,026	6,090	34.9	639,450
,425	876	5,288	6,090	32.6	487,200
,423	-	5,002	6,090	33.1	609,000
,446	1,714	5,644	6,090	40.5	578,550
,001	-	5,410	2,268	22.2	360,232
,390	4,223	5,936	2,268	25.5	170,100
,999	-	6,344	1,293	12.5	132,516
,189	-	5,980	334	5.5	48,784
,901	27,712	5,728	151	4.7	25,670
,392	-	5,152	151	2.4	16,520

A

TABLE XIII - Continued

		Loading *			
Item	Bearing	Axial	Radial (lb)	Moment (in.-lb)	Bear H L
13B	Tail Rotor and Access Bevel Ball Size 7026	755	2,053	1,849	7
14B	Generator, Drive Ball Size KD40CP, 2 on Shaft	- -	71 156	4 4	7 6
15B	Hydraulic Pump Drive Ball Size KD40CP (Kaydon)	-	53	-	12
16B	Oil Pump Drive Ball Size 26 x 52 x 15	-	-	-	
17B	Planetary Support Ball Size KF65AH (Kaydon)	-	-	-	
18B	Freewheeling Ball Size KD45XP (Kaydon)	-	-	-	

* 65% - Items 1B through 10B
100% - Items 11B and 12B
75% - Items 13B through 15B
No calculations are made for items 16B through 18B

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13

TABLE XIII - Continued

ng * al)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
53	1,849	7,862	4,943	39.9	889,740
71	4	7,520 (1)	6,355	28.5	645,178
56	4	6,060 (2)		28.1	
53	-	12,732	3,813	19.7	387,107
	-	-	-	-	-
	-	-	-	-	-
	-	-	-	-	-

A

TABLE XIV. SUMMARY OF FUNC
TWIN 4800 HP, 3

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Veloc (ft/m
1G	Engine Gearbox Spiral Bevel Pinion	4800	8,797	7.75	67	2.5	9,000	20,37
2G	Engine Gearbox Spiral Bevel Gear	4800	8,797	7.75	99	2.5	6,091	20,37
3G	Input Spiral Bevel Pinion	4800	9,770	7.75	86	2.75	6,091	17,48
4G(A)	Input Spiral Bevel Gear	4800	9,770	7.75	231	2.75	2,268	17,48
(B)	Oil Pump Takeoff Spur	10	93	10.0	60	.50	2,268	3,56
5G	Lower Sun Gear	6400	9,607	5.186	48	2.70	2,268	-
6G	Lower Planet Pinion							
	With Sun Gear	6400	9,607	5.186	43	2.60	1,878	4,05
	With Ring Gear	6400	9,607	5.414	43	2.60	1,878	4,05
7G	Lower Ring Gear	6400	9,607	5.414	138	2.40	-	-
8G	Upper Sun Gear	6400	19,474	4.1586	48	4.10	585	-
9G	Upper Planet Pinion							
	With Sun Gear	6400	19,474	4.1586	43	3.90	485	1,31
	With Ring Gear	6400	19,474	4.3414	43	3.90	485	1,31

B

OF FUNCTIONAL GEAR DATA,
OO HP, 3000-AND 6000-HOUR TBO

Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EHD Film Thickness (in. x 10^{-6})	Average Power Loss	Number of Meshes
20,370	132	38,971	189,595	50.0	13.335	2
20,370	132	36,714	189,595	50.0	13.335	2
17,486	114	38,571	147,022	50.0	8.402	2
17,486	114	37,861	147,022	50.0	8.402	2
3,563	43	4,522	73,720	29.05	.032	1
-	70	35,895	151,619	28.5	1.272	4
4,055	70	39,279	151,619	28.5	1.272	-
4,055	30	32,784	135,723	29.5	1.313	-
-	30	48,852	135,723	29.5	1.313	4
-	70	38,612	157,868	16.5	1.285	6
1,319	70	41,589	157,868	16.5	1.285	-
1,319	31	37,837	138,893	17.1	1.402	-

A

TABLE XIV - Contin

Item No.	Item	HP	Tangential Load (lb)	Diametral Pitch	Number of Teeth	Face Width (in.)	RPM	Pitch Line Velocity (ft/min)
10G	Upper Ring Gear	6400	19,474	4.3414	138	3.70	-	-
11G	Tail Rotor & Acc. Drive Takeoff Spiral Bevel	1010	2,206	8.856	106	.70	4,943	15,490
12G	Gen. & Hydr. Pump Drive Takeoff Spur	65	184	8.0	72	.40	4,943	11,647
13G(A)	Gen. Drive Spur	65	184	8.0	56	.30	6,355	11,647
(B)	Hydr. Pump Drive Takeoff Spur	15	50	8.0	48	.30	6,355	9,982
14G	Hydr. Pump Drive Spur	10	50	8.0	80	.20	3,813	9,982
15G	Oil Pump Drive Spur	10	93	10.0	36	.25	3,780	3,563

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LE XIV - Continued

PM	Pitch Line Velocity (ft/min)	Temperature Rise (°F)	Bending Stress (psi)	Compressive Stress (psi)	EDH Film Thickness (in. x 10^{-6})	Average Power Loss	Number of Meshes
-	-	31	50,463	138,893	17.1	1.402	6
943	15,490	97	42,462	141,536	50.0	2.112	1
943	11,647	36	7,567	73,347	50.0	.074	1
355	11,647	36	10,021	73,347	50.0	.074	2
355	9,982	16	2,626	46,393	50.1	.020	2
313	9,982	16	3,734	46,393	50.1	.020	2
780	3,563	43	8,038	73,720	29,05	.032	1

A

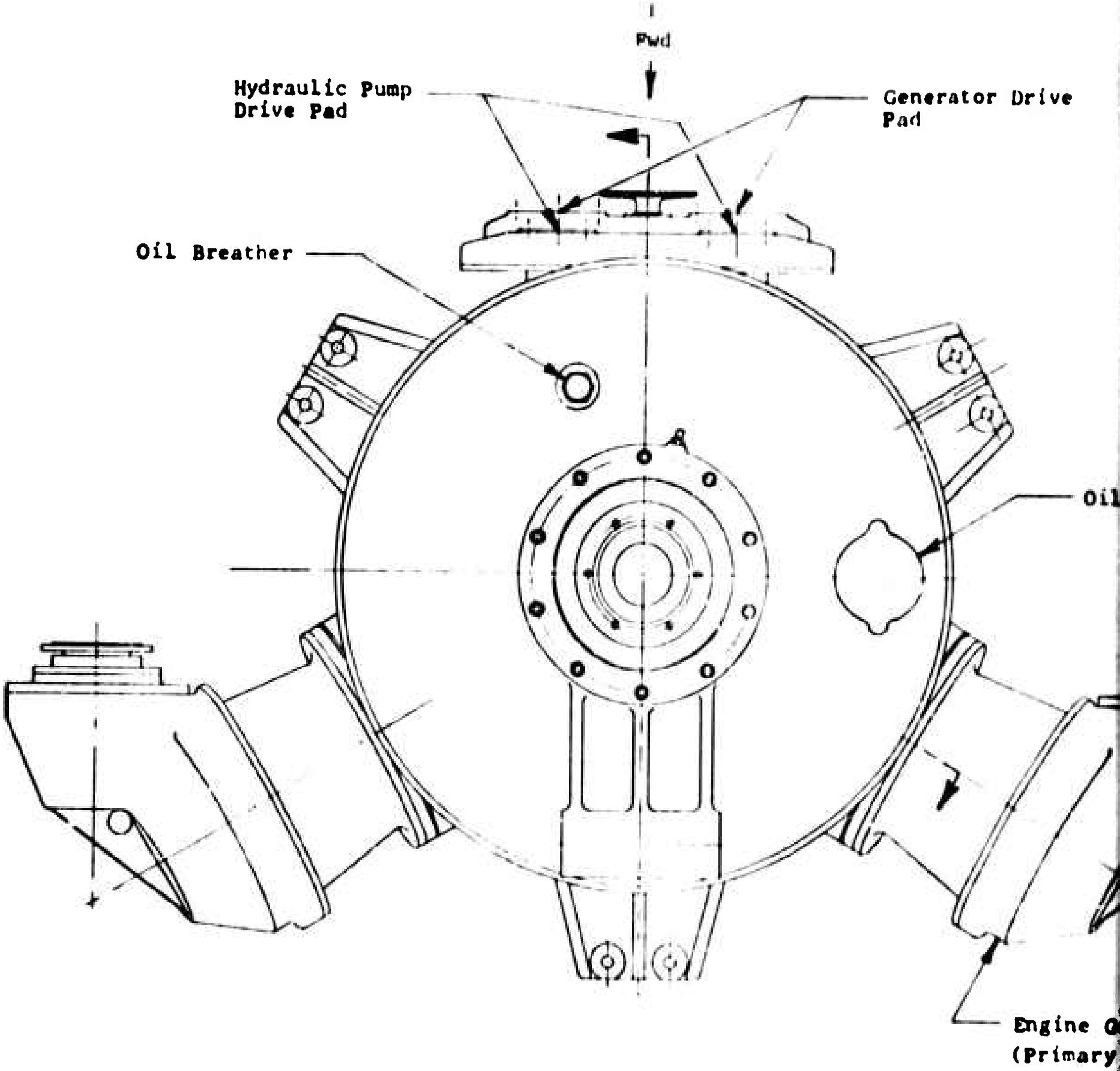

Figure 11. Twin 4800-HP Transmission, 6000-Hour TBO.

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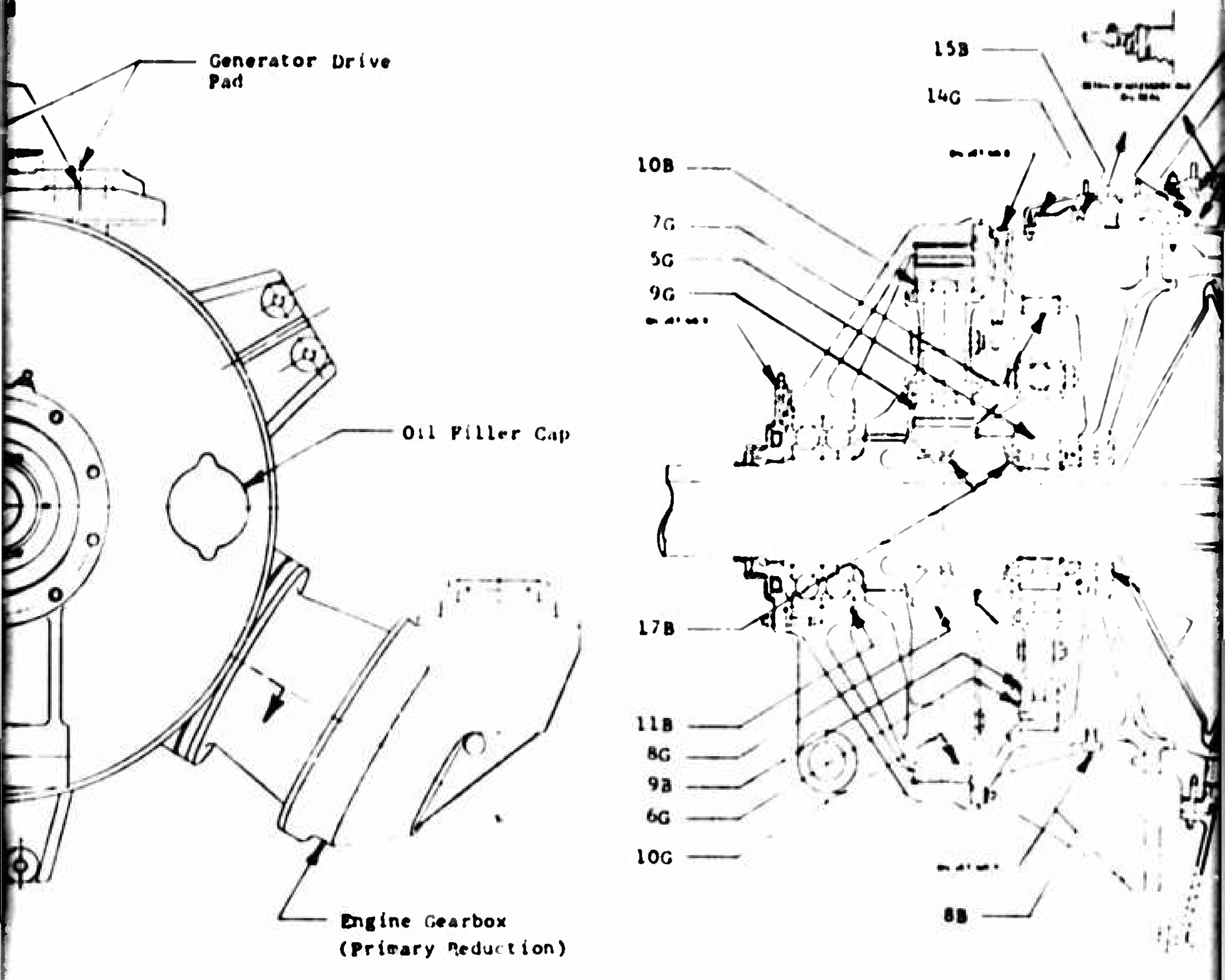
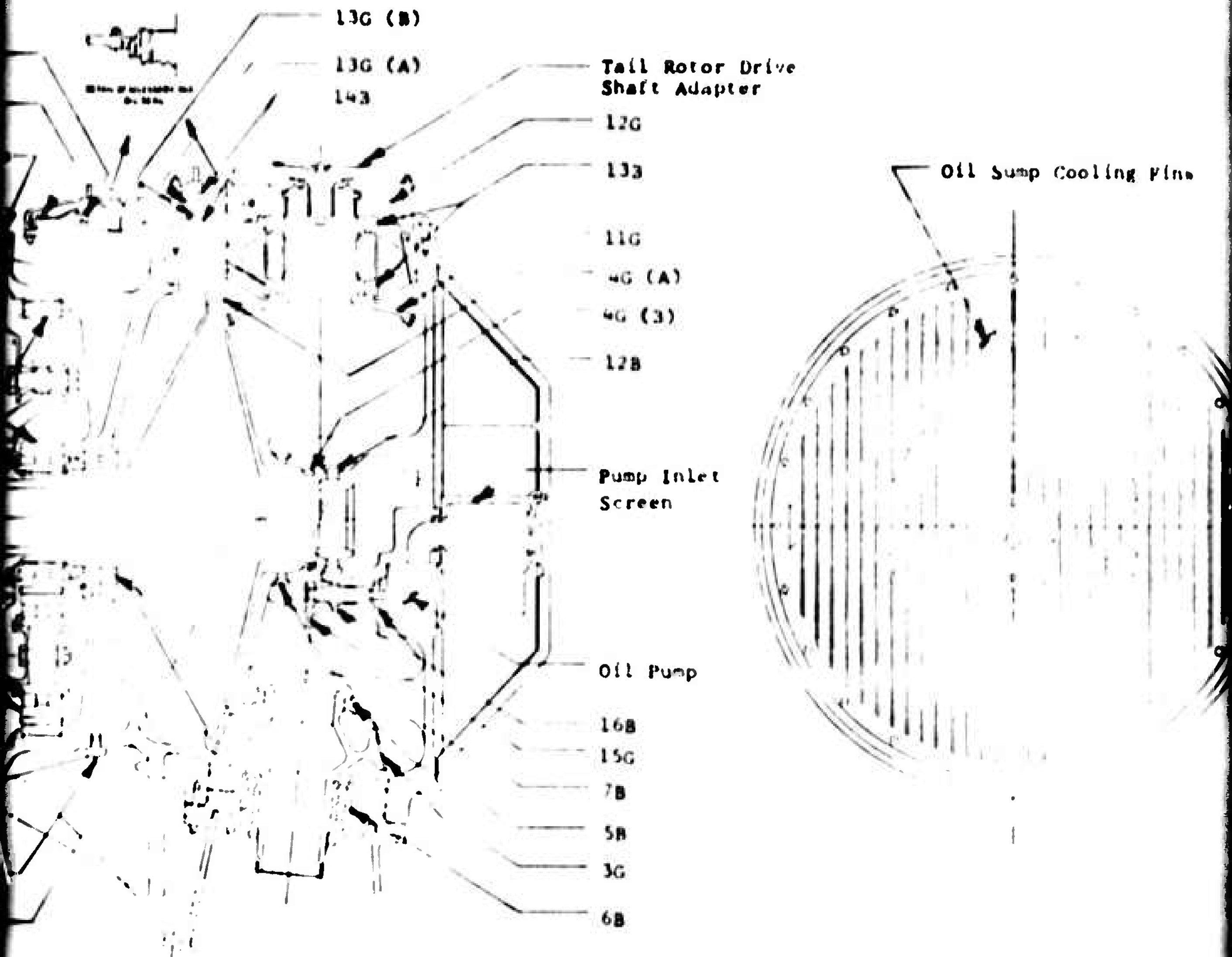
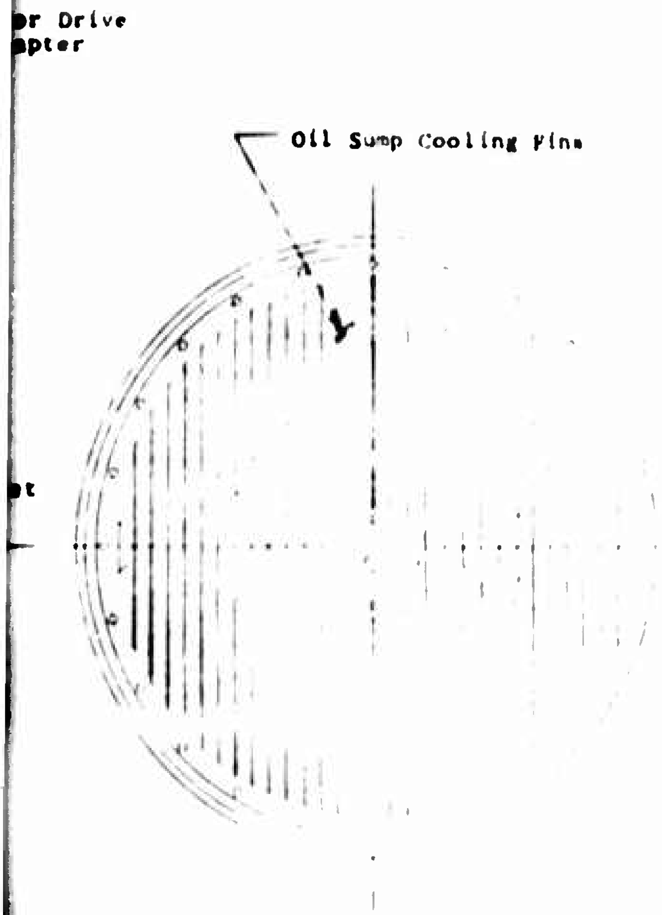
B


Generator Drive Pad
Oil Filler Cap
Engine Gearbox (Primary Reduction)
15B
14G
10B
7G
5G
9G
17B
11B
8G
9B
6G
10G
8B

r TBO.

C


13G (B)
13G (A)
14B
Tail Rotor Drive
Shaft Adapter
12G
13B
11G
4G (A)
4G (B)
12B
Pump Inlet
Screen
Oil Pump
16B
15G
7B
5B
3G
6B
Oil Sump Cooling Fins



GEARS

Item No.		Name
1G		Engine Grbx. Spiral Bevel Pinion
2G		Engine Grbx. Spiral Bevel Gear
3G		Input Spiral Bevel Pinion
4G	(A)	Input Spiral Bevel Gear
	(B)	Oil Pump Takeoff Spur
5G		Lower Sun Gear
6G		Lower Planet Pinion
7G		Lower Ring Gear
8G		Upper Sun Gear
9G		Upper Planet Pinion
10G		Upper Ring Gear
11G		Tail Rotor and Access Drive Takeoff Spiral Bevel
12G		Gen. and Hydr. Pump Drive Takeoff Spur
13G	(A)	Generator Drive Spur
	(B)	Hydr. Pump Drive Takeoff Spur
14G		Hydr. Pump Drive Spur
15G		Oil Pump Drive Spur

BEARINGS

Item No.	Name
1B	Engine Gearbox Pinion Roller Size 1020
2B	Engine Gearbox Pinion Ball Size 7218
3B	Engine Gearbox Gear Roller Size 1020

BEARINGS - Continued

Name

Engine Grbx. Spiral Bevel Pinion

Engine Grbx. Spiral Bevel Gear

Input Spiral Bevel Pinion

Input Spiral Bevel Gear
Oil Pump Takeoff Spur

Lower Sun Gear

Lower Planet Pinion

Lower Ring Gear

Upper Sun Gear

Upper Planet Pinion

Upper Ring Gear

Tail Rotor and Access Drive
Takeoff Spiral Bevel

Gen. and Hydr. Pump Drive
Takeoff Spur

Generator Drive Spur
Hydr. Pump Drive Takeoff Spur

Hydr. Pump Drive Spur

Oil Pump Drive Spur

Name

Engine Gearbox Pinion
Roller Size 1020

Engine Gearbox Pinion
Ball Size 7218

Engine Gearbox Gear
Roller Size 1020

Item No.	Name
4B	Engine Gearbox Gear Ball Size 7018
5B	Input Pinion Roller Size 1924
6B	Input Pinion Ball Size 7024
7B	Input Gear Roller Size 36 - 12.5 x 14 x 7.3425 PD
8B	Input Gear Ball Size 1932
9B	Lower Planet Roller Size 17 x 19 x 5.5707 PD 2 Rows: 19 Rollers/Row
10B	Upper Planet Roller Size 1/2 x .85 x 7.500 PD 2 Rows: 30 Rollers/Row
11B	Mast Ball Size 7234-9234
12B	Mast Roller Size 24 - 15 x 17 x 5.4925 PD
13B	Tail Rotor and Access Bevel Ball Size 7124
14B	Generator Drive Ball Size KD40CP (Kaydon)
15B	Hydr. Pump Drive Ball Size KD40CP (Kaydon)
16B	Oil Pump Drive Ball Size 26 x 52 x 15
17B	Planetary Support Ball Size AF65AH (Kaydon)
18B	Freewheeling Ball Size KD45XP (Kaydon)

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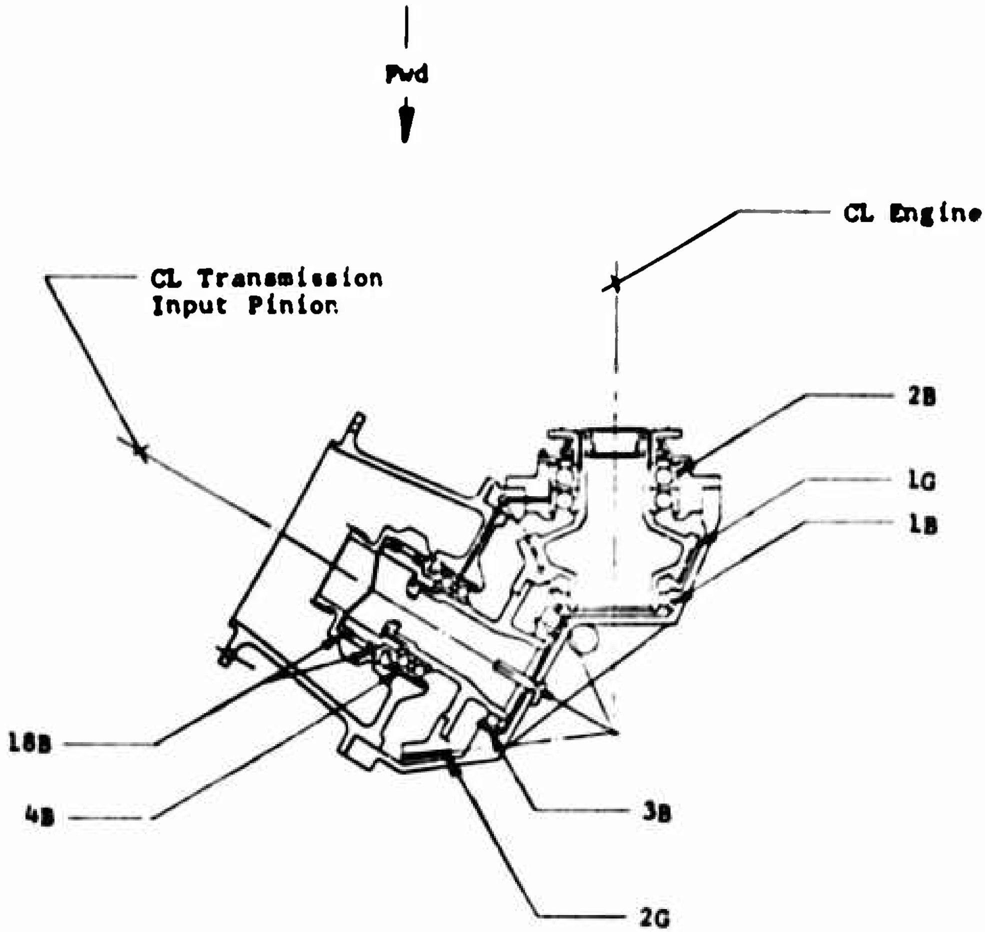

SECTION THRU INPUT REDUCTION GEARBOX

Figure 11 - Continued.

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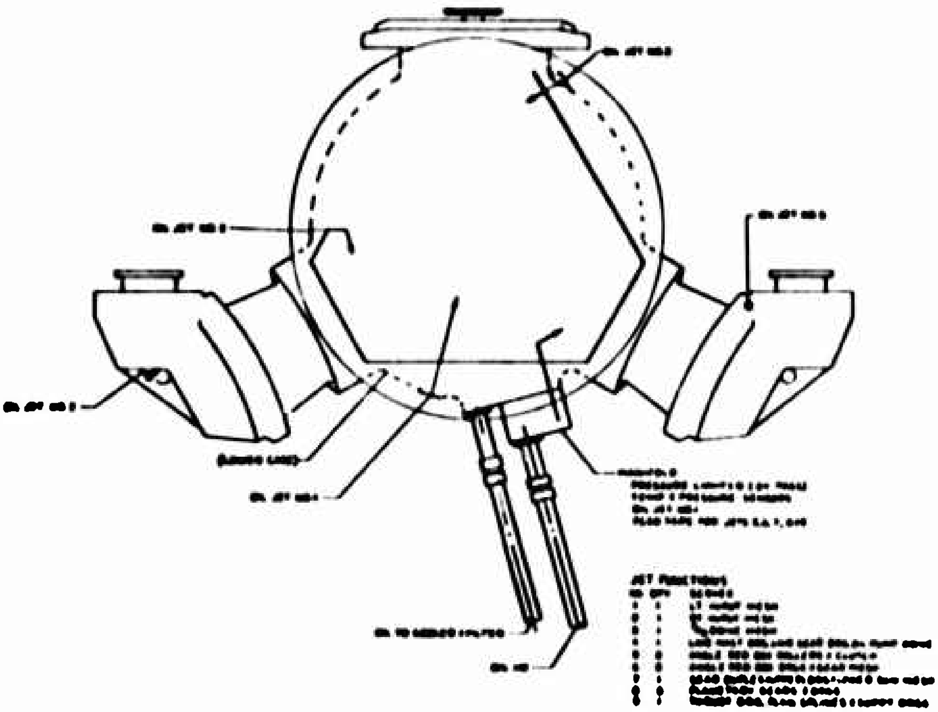
B

CL Engine

2B

1G

1B



LUBRICATION SCHEMATIC

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TABLE XV. SUMMARY OF BEARING LOADS AND LIVES

Item	Bearing	Loading * Axial	Radial (lb)	Moment (in.-lb)	Bea
1B	Engine Gearbox Pinion Roller Size 1020	-	3,495	-	1
2B	Engine Gearbox Pinion Ball Size 7128	1,717	2,029	1,543	1
3B	Engine Gearbox Gear Roller Size 1020	-	4,070	-	1
4B	Engine Gearbox Gear Ball Size 7018	724	1,367	862	1
5B	Input Pinion Roller Size 1020	-	3,275	-	1
6B	Input Pinion Ball Size 7024	2,036	1,487	2,497	1
7B	Input Gear Roller Size 36-12.5 x 14 x 7.3425 PD	-	4,992	-	1
8B	Input Gear Ball Size 1932	2,762	2,332	4,162	1
9B	Lower Planet Roller 2 Rows: 19 Rollers/Row Size 17 x 19 x 5.5707 PD	-	6,461	-	1
10B	Upper Planet Roller 2 Rows: 30 Rollers/Row Size 1/2 x .85 x 7.500 PD	-	10,610	-	1
11B	Mast Ball Size 7234-9234	25,065	9,385	27,296	1
12B	Mast Roller Size 24-15 x 17 x 5.4925 PD	-	12,306	-	1

B

NG LOADS AND LIVES - 6000-HOUR TBO, TWIN 4800-HP

*	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
6	-	19,664	9,000	43.6	900,000
9	1,543	11,696	9,000	48.99	810,000
0	-	11,000	6,090	32.6	609,000
7	862	12,352	6,090	32.2	548,100
5	-	14,744	6,090	35.7	730,800
7	2,497	11,096	6,090	42.2	730,800
2	-	12,344	2,268	20.9	365,960
2	4,162	13,344	2,268	24.7	362,880
1	-	12,960	1,293	11.7	138,853
0	-	13,368	334	5.1	55,102
5	27,296	18,864	151	5.4	25,670
6	-	12,288	151	2.3	16,520

A

TABLE XV - Contin

Item	Bearing	Axial	Loading * Radial (lb)	Moment (in.-lb)	Bearin Hou L1(
13B	Tail Rotor and Access Bevel Ball Size 7124	721	2,202	1,714	12,
14B	Generator Drive Ball Size KD40CP, 2 on Shaft (Kaydon)	- -	71 156	4 4	30, 24,
15B	Hydraulic Pump Drive Ball Size KD40CP (Kaydon)	-	53	-	12,
16B	Oil Pump Drive Ball Size 26 x 52 x 15	-	-	-	-
17B	Planetary Support Ball Size KF65AH (Kaydon)	-	-	-	-
18B	Freewheeling Ball Size KD45XP (Kaydon)	-	-	-	-

* 65% - Items 1B through 10B
100% - Items 11B and 12B
75% - Items 13B through 15B
No calculations are made for items 16B through 18B

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B

TABLE XV - Continued

ing * ial b)	Moment (in.-lb)	Bearing Life Hours L10 *	RPM	Minimum Film Thickness (10^{-6} in.)	DN
202	1,714	12,232	4,943	34.9	593,160
71	4	30,080 (1)	6,355	28.5	645,178
156	4	24,240 (2)		28.1	
53	-	12,732	3,813	19.7	387,107
-	-	-	-	-	-
-	-	-	-	-	-
-	-	-	-	-	-

TRADEOFF VARIABLES

TRANSMISSION WEIGHT

An assessment of the effect of an increase in the weight of the helicopter transmission in excess of "normal weight" to provide extended life is provided through a consideration of the attendant reduced operational flight envelope. There are immediately two ways in which the operational flight envelope can be affected by such an increase in weight, assuming that the helicopter has a developed operational flight envelope.

The increased transmission weight in one instance amounts to an increase in empty weight. By imposing a constant gross weight, the useful load or payload is reduced by the amount that the transmission increases in weight. In addition, further reductions in useful load are brought about by secondary structural weight increases associated with maintaining fixed crash-load requirements. Effects upon ship empty weight by component weight changes have been discussed extensively in Reference 13 for the medium-gross-weight helicopters. This in turn relates directly to the cost of operation of the helicopter, in that less cargo tonnage can be transported per unit time, fewer personnel can be transported per unit time, or fewer rounds can be loaded for an ordnance firing mission.

The statistical effect of reduced payload would be a function of the actual magnitude of transmission and system weight increase; however, the influence of the flight spectrum on gross weight distribution would not be large except where operation of the helicopter is at maximum gross weight primarily for every takeoff. The histogram of the gross weight spectrum would be shifted toward the high end by some amount depending upon the distribution mean and the weight increase. If operation is normally with somewhat less than maximum gross weight at takeoff, then the effect of reduced payload is nil. The approach to an assessment of the effect of reduction in payload would be to analyze a sampling of operational vehicles in various commands and determine gross weight histograms. The effect of reduced payload could then be directly related to actual operational requirements, and a relative cost factor could be determined for operation at reduced payload. This information is not presently available.

However, in the case of a new helicopter design, certain performance specifications must be met (such as hover at 4000 ft, 95°F), and the weight increase for extended transmission TBO could realistically reflect further increases in rotor diameter and weight. In order to develop a sufficiently general equation and yet retain a simplistic approach, the factor of 3:1

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on total transmission weight increase will be used for assessing the quantity of fuel consumed for cost purposes only. The approach used to derive the original incremental penalty will simply be to trade the increased weight for a reduction in fuel quantity. The reduced operational flight envelope would then be affected by endurance or range restriction. As with the reduced payload, the gross weight would be constant, but in this case the payload would also be constant. The tradeoff for the weight variable would be extended life obtained by increased transmission weight at the expense of reduced range or endurance.

The basis for the Δ-weight/Δ-endurance tradeoff is predicted on the following assumptions:

1. Vehicle gross weight is constant for all TBO levels.
2. Vehicle payload is constant for all TBO levels.
3. Direct operational costs will be based on fuel costs in the RVN theater.
4. An average specific fuel consumption is used for all three power levels. This is to be 0.70 lb/hr/hp.

Table XVI reflects certain existing trends in turbine-powered helicopter design and operational characteristics. The information was extracted from detail specifications for each model.

The formulas used to equate weight increase to additional operating cost are:

1. $\Delta C_o = (\Delta T)(C_o)$
2. $\Delta T = (\Delta W)/(SFC)(P_m)$

ΔC_o = additional cost of operation, \$/hr

ΔT = loss of flight time/mission, hr

C_o = cost of operation, \$/hr

ΔW = weight increase for extended TBO, lb

SFC = specific fuel consumption, lb/hr/hp

P_m = mean mast power

Credibility of the above formulation is advanced by the information in Reference 1, wherein histograms of flight lengths and

flying mission lengths are shown. If it were assumed that every flight would be as long as maximum endurance predicated on fuel quantity, then the ΔC_o above would be inconsistent with actual operating cost increase. The required refueling time and lost motion involved would then be effective in the increased cost figure. However, as can be seen in Figures 12 and 13, the flight lengths and flying mission lengths are considerably less than maximum endurance. Therefore, the required additional refueling times are very few, since the relative number of flying mission times above 3 hours are fewer than 0.1%.

The effect of increased transmission weight on cost is determined by assessing the cost of fuel load required to satisfy the conditions of preserved range or preserved endurance. The specific fuel cost in the RVN theater is the basis for this analysis. This has been determined to be \$0.016 per pound. On that basis, C_o is determined as follows:

$$C_o = (0.016)(SFC)(HP_{AVG}) \text{ (Reference Parametric Discussion)}$$

$$= (.016)(.7)(.65)(667) = \$4.80/\text{hour (twin 500)}$$

$$= (.016)(.7)(.65)(2000) = \$14.50/\text{hour (twin 1500)}$$

$$= (.016)(.7)(.65)(6400) = \$46.5/\text{hour (twin 4800)}$$

For an average mission endurance of 3.37 hours (Table XVI) Formula 1 reduces to:

$$3.\quad \Delta C_o = \frac{.016}{3.37}\Delta W_T \quad (\Delta W_T = \text{Transmission Weight Increase})$$

$$= .0046\,\Delta W_T$$

An assessment of the penalty incurred for overweight poundage is provided through the use of the previously discussed 3 to 1 multiplication factor (see page 111). As previously noted, this factor is used to account for the additional power required for normal flight, additional fuel weight, additional structure for fuel capacity, ballast, etc. The cost per flight hour figures shown in Table XVII are, therefore, arrived at by finally modifying equation 3 above as:

$$4.\quad \Delta \text{Cost} = (3)(.0046)\,\Delta W_T$$

$$= .0138\,\Delta W_T$$

The results of the weight and cost analysis are shown in Table XVII.

Average 43 Minutes

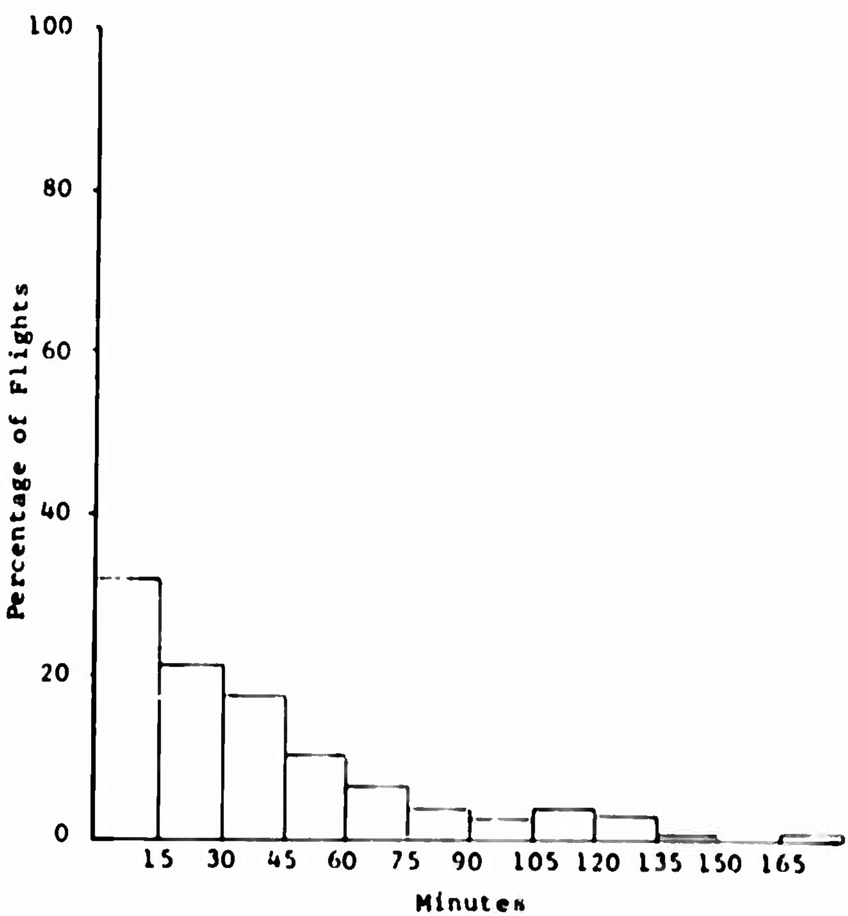

Figure 12. Model AH-1G Flight Lengths.

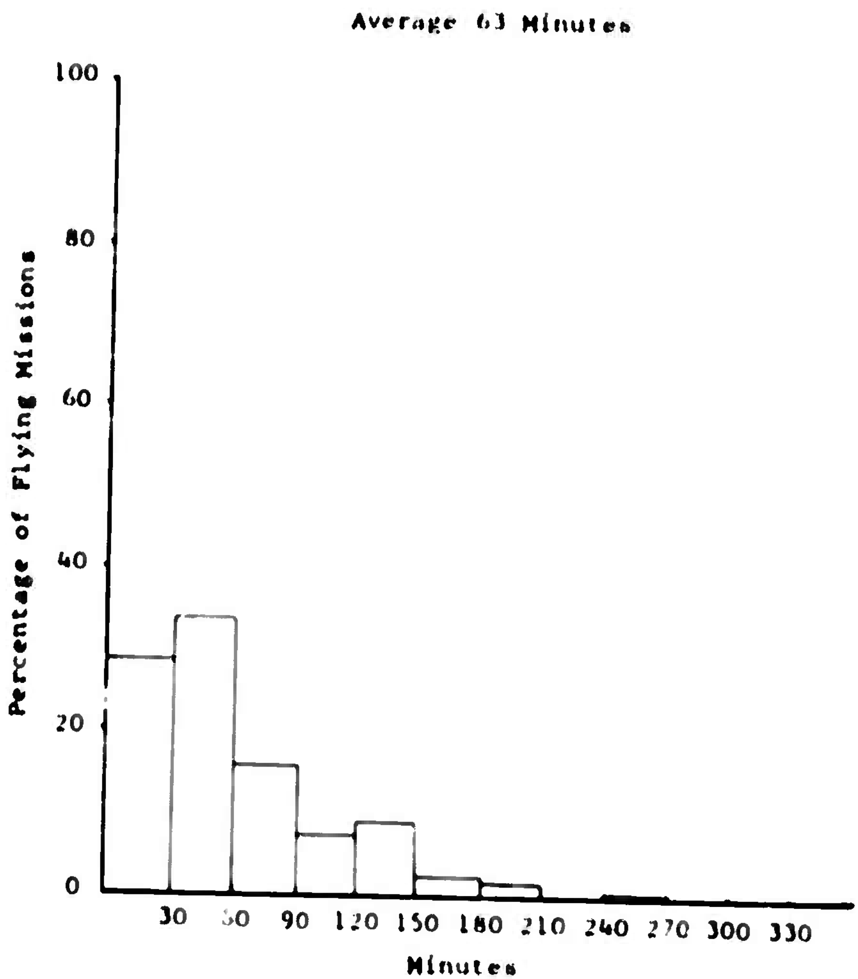

Figure 13. Model AH-1G Flying Mission Lengths.

TABLE XVI. TYPICAL HELICOPTER DESIGN AND OPERATIONAL CHARACTERISTICS

Model	Fuel Capacity (lb)	SFC (lb/hr/hp)	Range (n mi)	Estimated Endurance (hr)	Fuel Reserve (lb)
UH-1A	1072	0.719	220	3.0	107.2
UH-1B	1592	0.630	324	-	159.0
UH-1C	1600	0.598	322	3.35	160.0
UH-1D	1230	0.702	204	3.6	1/2 hr
UH-1E	1230	.600	204	3.4	1/2 hr
UH-1F	455	.706	260	3.5	46.0
UH-1G	1381	.592	260	-	138.0
Mean	14.4% GW	.656	248	3.37	10% Fuel Capacity

TABLE XVII. TRANSMISSION WEIGHT AND ΔCOST

	1200 TBO Trans. Weight (lb)	3000 TBO Trans. Weight (lb)	ΔW_T	ΔCost \$/flt hr	6000 TBO Trans. Weight (lb)	ΔW_T	ΔCost \$/flt hr
Twin 500 HP	220	259	39	.48	240	20	.62
Twin 1500 HP	618	701	83	1.14	686	70	.99
Twin 4800 HP	2582	2874	292	4.04	2786	204	2.62

SIGNIFICANCE OF SPECIFIED GROSS WEIGHT AND RANGE

Assumptions:

1. Constant gross weight

. Reduced fuel capacity or reduced payload

The specification endurance of most helicopters used in the RVN theater is approximately 3.5 hours with a specific fuel consumption of .6 to .7 lb hr hp (Table XVI). In the specification of fuel load, endurance, and range, there is a fuel portion allotted for warm-up and preflight hover and another portion for reserve. The range then is a function of the remaining fuel capacity and provides from 190 to 325 miles range without the use of the reserve fuel.

In Reference 1 it was shown that for three Bell helicopters the total operation time was divided into averages of 27.5% ground time and 72.5% flight time. Also, it was shown that the AH-1G and UH-1H flight lengths averaged 43 minutes and 26 minutes, respectively; flying mission lengths averaged 63 minutes and 71 minutes, respectively. From these statistics, it can be shown that a very appreciable amount of military helicopter operation time the RVN theater is spent in ground operation or in hover maneuvers.

Since only about one-third of the flight time capability is used in flight operation, the significance of a reduction in fuel capacity of as much as 10% (this would generally amount to 100 lb) is slight. For example, this 10% decrease would amount to approximately 30 minutes of cruise on the UH-1H and would reduce the endurance to approximately 3 hours. In the flying mission histogram (Figure 13), it was shown that only about 1% of the flight spectrum was devoted to operation in excess of 3.5 hours. Thus, if an extended TBO period were stipulated by design specification, the extended TBO could be accomplished at the expense of a reduction in fuel capacity without affecting over 90% of the current missions.

In order of significance, the gross weight spectrum next enters into the effect of reduced fuel capacity. If the relative importance is assumed to be greater for range than for payload, then the gross weight would be constant, the payload reduced an amount, and the fuel capacity increased a like amount. Justification for this deviation from the present specification would arise from a spectrum wherein the percentage of time at maximum gross weight would be expected to be small, as was the percentage of time in maximum endurance flights as seen above.

In either event, in terms of actual dollar expenditure for materials and service, the extended TBO design would become immediately desirable in a tradeoff of either vehicle range or payload, when neither of the latter is significantly exploited. Reduction in logistics, transportation costs, and overhaul costs attained by fewer overhaul cycles and less actual hardware constitutes a real expenditure decrease; while the loss or shrinking of an operating envelope that is seldom used in the limits would constitute an imaginary expenditure increase.

REDUCED ALLOWABLES

Transient or brief operating periods at power levels above the maximum design power level occur depending upon design criteria, locale, and circumstances.

During such overpower periods, the gear and bearing bulk temperatures may increase under the influence of the additional friction losses. These higher temperatures serve to lower lubricant viscosity in the load conjunctions, thus reducing elastohydrodynamic (EHD) film thickness at these contacts. For a given surface roughness, the ratio of film thickness to composite roughness of the contacting bodies is reduced. This, in turn, may additionally increase the actual peak stresses in these contacts (above that due simply to the incremental overpower load increase) either due to increased traction forces or simply more direct and less cushioned contact between the asperities of the given roughness. These peak stress values frequently initiate surface fatigue or cracking, which may hydraulically propagate in the thin synthetic oil environment or produce gear scuffing or bearing raceway smearing, depending upon the energy density in the contact.

Chemical effects in the conjunction environment have also been shown to influence failure mode and propagation rate in these near-boundary-layer lubrication regimes (Reference 12).

The gear pitting fatigue failure mode occurs most frequently in the low pitch-line velocity - final rotor drive portions of the transmission. However, in low-speed meshes, the stress cycle accumulation rate at overpower conditions may not cause premature failure at the 1200-hour TBO level. With an extended TBO period, cycle accumulation can attain a damaging level sufficient to cause premature failure.

A study of the power spectrum of the AH-1G (Reference 1) was made to determine a representative overpower frequency. The transmission used in this helicopter attains an MTBR of 90%

of its scheduled 1100-hour TBO (Reference 2). It is powered by a 1400-hp sea-level standard-day rated turbine engine. The helicopter is equipped with a transmission system rated for operation at 1250 maximum, with corresponding torquemeter red-line markings in the cockpit.

The power spectrum analysis on monitored AH-1G helicopters in the RVN theater (Figure 2) reveals that 1.567% operational time was spent above 1100 hp with .007% at the engine rating of 1400 hp. For the 1100-hour TBO, this helicopter transmission sees 17.27 hours above design power and .08 hour at engine rating.

While this represents but 5.5 million cycles at elevated stresses on the final stage sun gear, the frequent replacement of these parts at overhaul (Reference 2) supports the contention that the wear condition would be intolerable at an increase of three or six times cyclic exposure in the extended life transmission. Consequently, in the design of the extended life transmission, it is necessary to employ a reduction factor for establishing safe operating stresses. This factor was set equal to the ratio of power available to the drive system design power:

$$\text{Overpower Factor} = \frac{\text{Power Available}}{\text{Design Power}}$$

$$= \frac{1400\ \text{HP}}{1250\ \text{HP}} = 112\%$$

This factor is applicable only to the planetary reduction gears, since the primary reduction stages are designed for 100% single-engine power. The design allowable for compressive stress of 170,000 psi (assuming well-aligned parallel gear meshes with minimized stress concentration at tooth ends) was established on the 1200-hour TBO design, and the 1.12 factor was applied to the 3000-hour and 6000-hour designs. Since the idealized compressive stress varies as the square root of the power level (for constant speed), the design allowable stress at normal rating was thereby decreased to 160,000 psi.

As previously discussed, the true operating stresses probably increase substantially faster than the idealized square root relationship (accounted for by the 1.12 factor) in conjunctions where the ratio of EHD film thickness to composite roughness is less than unity. Therefore, the extended-life design must embody additional improvements in these areas. The effects of misaligned contacts can never be properly assessed in initial design. However, measures to improve the composite surface roughnesses in the conjunctions and to limit the overload

stress concentrations can and must be taken (Reference 12). Final honing of the gear teeth satisfies the former requirement, and the latter can be fulfilled by using tooth crowning (or end relief), adequate involute profile modifications to promote load sharing between adjacent tooth pairs, increased compliance in the planetary carrier structure and ring gear support to assure load sharing among the various planet idler gears, and relatively compliant bearing outer ring support structure to permit outer ring conformance to the load with an attendant increase in the number of rolling elements in the load zone.

The application of the 1.12 load factor results in increased weight of gear and bearing elements, while the referenced measures to refine surface roughness and promote load-limiting geometric conformity in the conjunctions induce higher manufacturing costs by virtue of the additional process time involved. The costs associated with the weight increase are relatively constant over the useful life of the transmission. The additional cost factors used to produce better surface finishes are related to machining processes that are subject to future cost reduction as they become more widely used. Although this potential reduction was not assumed for purposes of this study, it does represent an area of future cost reduction.

MACHINING AND PROCESSING COSTS FOR EXTENDING TBO

Processing variable controls requisite to attainment of high gear life are associated with active profile surface texture and topography. Planetary gear profiles in the UH-1 planetary assemblies are manufactured according to normal aircraft gear grinding and finishing techniques. Surface finishes are 20AA or better, waviness requirements are within .000070 inch, and profile errors are within .0002 inch. The lower planetary assembly operates satisfactorily at 1100 hp with no distress at a calculated surface compressive stress of 156,000 psi. These same planetary idlers have been tested satisfactorily at 2200 hp at a calculated surface compressive stress of 170,200 psi without distress. Lead misalignment between sun and planet or ring and planet gear teeth is precluded by the carrier design; hence, there is no end overloading in these meshes due to misalignment. However, the ring gear face width is considerably less than the planet gear face width, and end effects are present. These are similar to the end loading effects in uncrowned cylindrical roller bearings. The actual load condition produces a peak compressive stress at the ends of the ring gear teeth that is much higher than the nominal calculated stress. Such a stress distribution is quite satisfactory for the UH-1 TBO period of 1100 hours, and perhaps to 2000 hours.

However, to extend the life to 3000 or 6000 hours requires a reduction of these end effects that will reduce the peak stresses to the nominal values for a more even axial load distribution. The upper planetary assembly in the UH-1 does not have ideal alignment in the gear teeth under all load conditions. The carrier deflects under load in such a manner that the upper end of the sun gear teeth would absorb more load than the nominal calculated gear load. Helix or lead slope correction to provide parallel alignment at nominal values of 900 hp is used. At higher power levels, the upper ends of the teeth operate under a higher stress than occurs with an evenly distributed load. Again, TBO could not be extended significantly under these operating conditions (Reference 2). However, test work reported in Reference 12 reveals that under the very thin film lubrication regimes occurring in this mesh life (as defined by the pitting failure mode) is relatively insensitive to load distribution alone. It becomes necessary to change lubrication states and to use differential hardness materials, in addition to improving load distribution.

Modification of the basic design to include geometric refinements that will allow longer gear life requires additional machine operations in manufacture. Additional machining is in the nature of honing, crowning, and lead modification, which normally would not be accomplished on a gear for a low TBO gearbox design. In this instance, the lubrication state is changed by surface finish improvement rather than by a lubricant viscosity change since the lubricant is specified.

An average cost figure based on the incorporation of the above refinements in recent production UH-1 gearboxes is used as an additional processing factor for the extended TBO gearboxes. This figure is set at $200 for the upper and lower planetary assemblies used in the UH-1 main transmissions. This value is used in determining the additional processing costs of the planetary gears in the extended life transmissions. For the incremental increase in cost attributable to additional processing, it is assumed that this cost increase is a linear function of horsepower. Figure 14 shows that a linear relationship exists between horsepower and weight. Therefore, since larger and heavier gears normally require greater processing time than smaller counterparts, the assumption of linearity between additional processing cost and horsepower is sufficiently accurate. The respective additional manufacturing costs shown below are determined by:

$$\Delta \text{Cost} = \left(\frac{HP_X}{HP_{UH-1H}}\right) \frac{\text{No. Planetary Assys} \times \$200}{2}$$

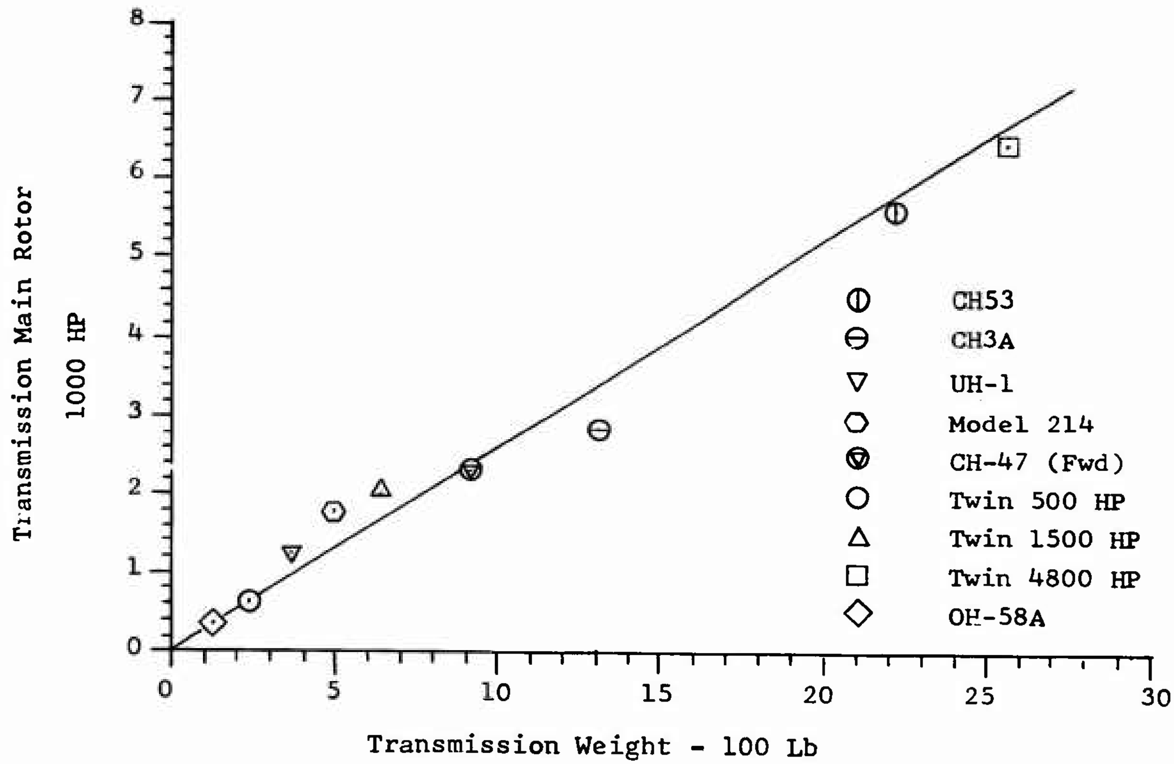

Figure 14. Transmission Weight Vs Horsepower.

Transmission	TBO	Δ Cost, $	Cost, $/ Flt Hour
Twin 500	1200	-	-
	3000	60	.02
	6000	60	.01
Twin 1500	1200	-	-
	3000	364	.121
	6000	364	.06
Twin 4800	1200	-	-
	3000	1160	.388
	6000	1160	.194

MATERIAL COST FACTORS

The changes required to attain a 3000-hour TBO evolve only from higher bearing and gear life requirements. The materials used in the 3000-hour TBO designs are the same as the 1200-hour designs, but bearing and gear sizes are larger. No material cost changes were incurred in changing certain cases from magnesium to aluminum. The bearing sizes in the 6000-hour TBO transmission are the same as the 1200-hour TBO transmission, but the material is conformally forged consumable electrode vacuum-arc-remelt AISI M-50 steel rather than degassed AISI 52100 steel. The following estimates are made for bearing costs of the nine transmissions considered in this study:

Transmission	TBO	Total Bearing Cost $	Δ Cost, $	Additional Cost $/ Flt Hr
Twin 500	1200	213	-	-
	3000	265	52	.017
	6000	638	425	.071
Twin 1500	1200	1840	-	-
	3000	2300	460	.153
	6000	5620	3780	1.26
Twin 4800	1200	4680	-	-
	3000	5850	1170	.39
	6000	14100	9420	1.57

TRANSMISSION EXCHANGE DOWN TIME AND MAINTENANCE

The effects of down time for gearbox change are not considered in this report, since documented information was not obtainable. The temporary loss of the helicopter for gearbox change,

however, can represent a very sizeable cost as well as present a hazard to the safety of a combat-operational command. It is conjectured that the actual costs of removing and replacing a gearbox would be minimal, while the lag in supplying the gearbox to the helicopter could amount to a time interval sufficient to entirely override the cost effectiveness of any developed TBO. However, this omission serves only to lend conservatism to the final study results.

Periodic maintenance for the extended-life transmission is to be the same as presently required on UH-1 helicopters in the RVN theater. Specifically, the transmission periodic maintenance (25-hour interval) is as follows: Inspect transmission and connections for damage and oil leaks. Check sump for water contamination and for oil level. Check transmission oil filter for bypass indication if so equipped. Inspect and clean transmission oil screen and magnetic plug. On electrical chip detector, check continuity.

OVERHAUL AND TRANSPORTATION COSTS

The cost of overhaul and transportation of the nine transmissions developed in this study is based on the costs reported from U. S. Army Aeronautical Depot Maintenance Center in Corpus Christi, Texas, and results of a study performed by Sikorsky Aircraft for U. S. Army Aviation Systems Command, St. Louis, Missouri (contract DAAJ01-68-C-1395(c)).

Overhaul and transportation costs of seven transmissions (UH-1, Vertol CH-47 forward, aft, and combining gearboxes, and Sikorsky CH3A, CH3C, and CH53) were analyzed, with results shown in Table XVIII. The quantity of transmissions reviewed in a 5-month period as reported in Reference 2 (173 UH-1's, 40 CH-47's forward, 32 CH-47's aft, and 5 CH-47's combining) in conjunction with their respective weights and overhaul costs indicates that overhaul costs could generally be predicted as shown in Figure 15. Transportation costs as shown in Table XVIII indicate that transmission weight and size are instrumental in determining general transportation costs. It is not expected that quantity of transmissions would appreciably influence the transportation costs. The curve shown in Figure 16 is based on the information in Table XVIII and the respective weights of the transmissions shown. Transportation costs of the nine transmissions developed in this study were then based on this curve. Further justification is given below, wherein very high MTBR's would be attained.

It is anticipated that the extended-life transmission would not incur part replacement frequency as high as the low TBO gearboxes. Those areas of wear that are normally tolerable

TABLE XVIII. OVERHAUL COST BREAKDOWN

Transmission	Turnaround OH Cost ($)	Transportation RVN Round Trip ($)	OH Period (hr)	MTBR (hr)	Cost * ($/flt hr)
UH-1	2300	610	1100	995	2.93
CH-47B Aft	6362	1410	600	523	14.80
CH-47B Fwd	5909	1530 (Avg)	600	449	16.60
CH-47B Comb	6478	620	1200	1189	5.95
CH3A	9100	2080	850	345	32.30
CH3C	9100	2080	500	286	38.90
CH53	15505	3655	500	204	94.00

$$* \text{ Cost/Flt Hr} = \frac{(\text{OH} + \text{Transport})}{\text{MTBR}}$$

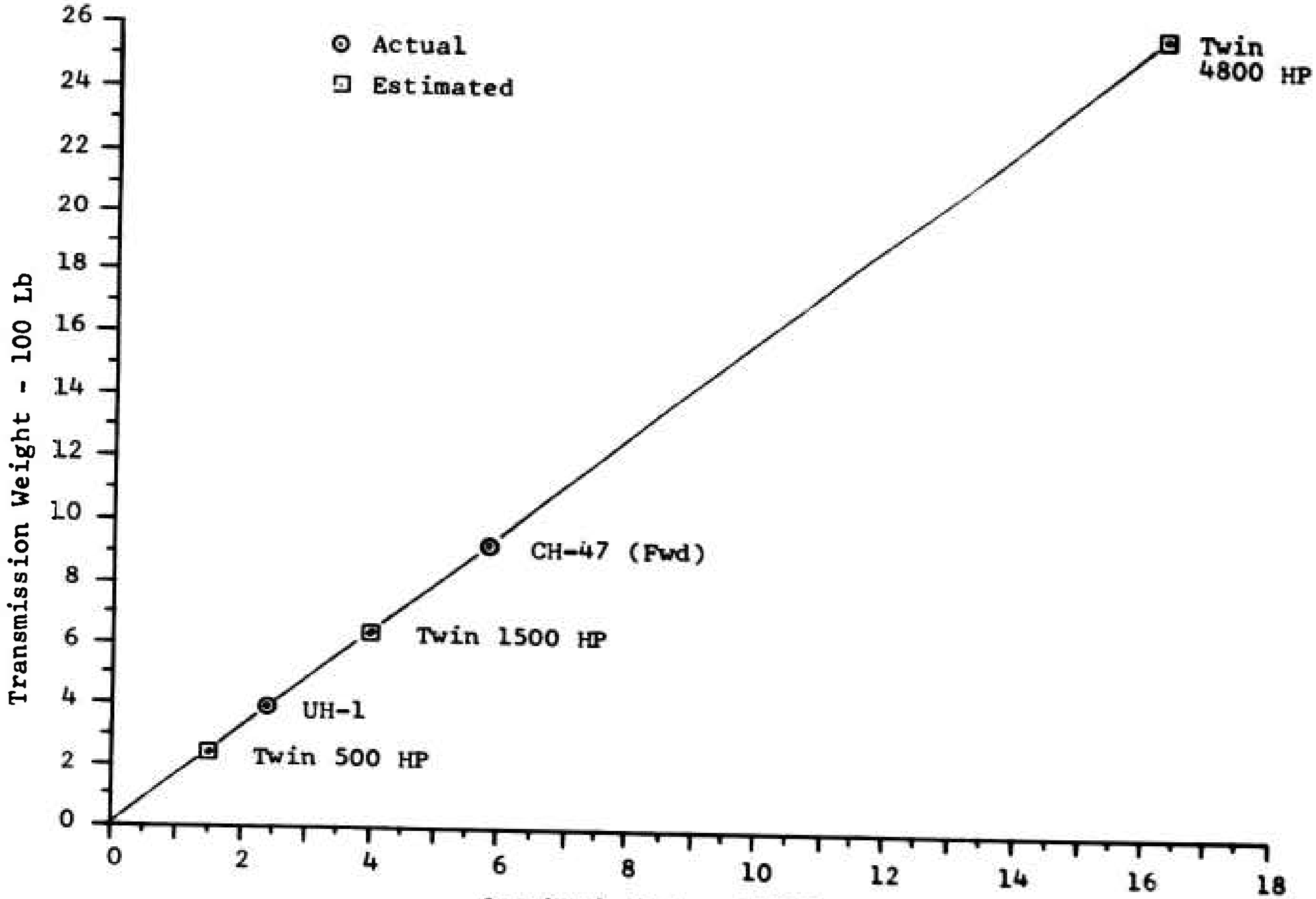

Figure 15. Overhaul Cost Vs Transmission Weight.

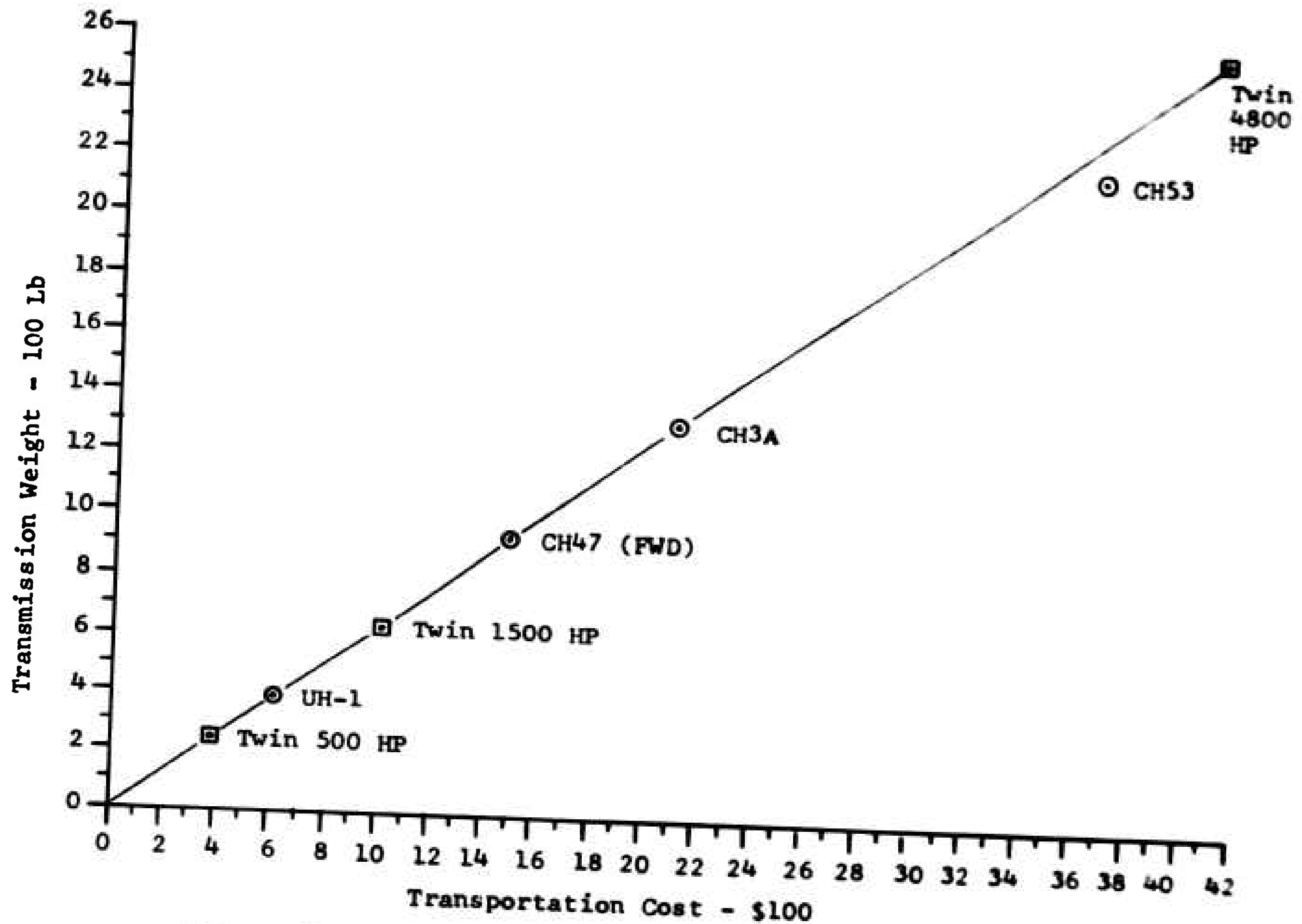

Figure 16. Transportation Cost Vs Transmission Weight.

for low TBO, such as wear due to bearing inner ring creep on the shaft, bearing spacer wear, and gear profile wear, are specifically designed so as to be eliminated from the extended life transmissions. In like manner, the allowable stresses are low enough to prevent spalling fatigue failure in gears and bearings.

Bearing ring creep is eliminated by reducing gear reaction loads through use of larger gears and by proportioning gear shaft and bearing diameters to provide interference fits sufficient to preclude creep even at overtorque conditions. Bearing spacer wear is normally eliminated when ring creep is eliminated.

Gear profile wear is eliminated by virtue of reduced allowables. Under normal operating conditions, the meshing gears are separated by an appreciable lubricant film, so asperity contacts (wear) are nonexistent. At overtorque conditions, the transmitted gear loads are increased and the resulting thermal rise is equivalent to that normally experienced at the stabilized operating level of the low TBO designs. Random asperity contact may be made, but wear is not significant since overtorque duration is low.

The overhaul costs reflected in Figure 15 would then represent probable upper limit for transmissions designed for extended life.

RELIABILITY

Gear and bearing design data and functional data are tabulated for each transmission and presented after each respective transmission layout. The information included in the tables was used to calculate generic failure rates of both the gears and the bearings (Tables XIX and XX).

Gear pitting failure rates were determined from composite life calculations based on a dispersion exponent of 5.5. The resulting composite pitting lives, while short of TBO in each case, reflect the increase in life obtained through design to reduced allowable stresses. The pitting life of each gear was calculated at 100% design load, while the cubic mean power level was previously determined to be 65% of maximum design. No calculations were made at the mean power level. However, since the gears are not treated in the same statistical manner as the bearings, the relative reliability indexes given will suffice for comparing the extended life designs with the base line design.

Generic failure rates are shown for bearings in each transmission, and the progressively extended TBO transmissions exhibit lower characteristic failure rates. The L_{10} TBO design life ratio requirements of the base line 1200-hour transmissions are considerably higher than the respective 3000-hour and 6000-hour transmissions, yet the failure rates are inherently lower for extended life. The tabulated rates suggest an increased reliability of the extended life transmissions despite the less stringent relative L_{10} requirements.

Considering the poor basic correlation shown between calculated bearing lives and observed failures reported in Reference 2, there exists little justification for further statistical treatment.

Validation of the cost per flight hour data presented in this study requires the tacit assumption that the ratio of mean time between rework (MTBR) to TBO is constant at all three TBO design levels. The real cost per flight hour for a given transmission is strongly influenced by MTBR. Increasing TBO without increasing the transmission reliability proportionately would not only result in an unsatisfactory maintainability, but would also increase the relative cost per flight hour values for the extended life versions above the values shown. For simplicity of the study, the MTBR is considered to be coincident to the TBO. Design measures have been taken to provide increased life at a constant reliability level for each increased TBO design in order to ensure this desired proportionality. While the resulting absolute dollar value rates may be optimistic, their relative values should remain accurate.

TABLE XIX. CUMULATIVE GEAR LIVES AND GENERIC FAILURE RATE

TRANSMISSION	CUMULATIVE LIFE L_s	FAILURE RATE TBO/L_s	FAILURE RATE 1000/L_s
Twin 500 HP			
1200-Hour TBO	366	3.279	2.732
3000-Hour TBO	419	7.160	2.387
6000-Hour TBO	419	14.320	2.387
Twin 1500 HP			
1200-Hour TBO	464	2.586	2.155
3000-Hour TBO	497	6.036	2.012
6000-Hour TBO	497	12.072	2.012
Twin 4800 HP			
1200-Hour TBO	1122	1.070	.891
3000-Hour TBO	1146	2.618	.873
6000-Hour TBO	1146	5.236	.873

TABLE XX. COMPOSITE BEARING LIFE AND GENERIC FAILURE RATES

Transmission		Composite Life L_{10}	L_{50}	Failure Rate TBO/L_{50}	1000/L_{50}
Twin 500	1200-Hr TBO	1926	3100	.3871	.3225
	3000-Hr TBO	3338	5375	.5582	.1861
	6000-Hr TBO	7390	11898	.5043	.1353
Twin 1500	1200-Hr TBO	1591	2561	.4686	.3905
	3000-Hr TBO	2457	3955	.7585	.2528
	6000-Hr TBO	6314	10165	.5902	.0984
Twin 4800	1200-Hr TBO	1335	2150	.5582	.4652
	3000-Hr TBO	2529	4071	.7369	.2456
	6000-Hr TBO	5262	8471	.7083	.1180

CONCLUSIONS

On the basis of overhaul and transportation costs as well as fuel cost in the RVN theater, the tradeoff study shows that the 3000-hour TBO design and the 6000-hour design are cost effective. As shown in Figure 17, the extended life designs are cost effective at all power levels with relatively decreasing cost of operation as a function of increasing time between overhaul.

Final design optimization of a transmission would be attained through the use of conventional bearing materials (SAE 52100 steel) in certain locations that do not critically demand extremely high endurance characteristics obtained from the more expensive M-50 steel bearings. The hybrid use of these less expensive bearings would result in the most cost effective TBO transmission. In summary:

1. Extended-life transmission design is feasible and cost effective.

2. Design optimization would provide a maximum overhaul life commensurate with minimum cost.

3. Increased life-transmission design provides equal or better reliability with lower failure rates than low-life counterparts.

TABLE XXI. EXTENDED-LIFE TRANSMISSION COST, $/FLT HR

Cost Item	TBO, Twin 500			TBO, Twin 1500			TBO, Twin 4800		
	1200	3000	6000	1200	3000	6000	1200	3000	6000
Weight	0	.48	.27	0	1.14	.97	0	4.04	2.82
Efficiency	0	.0041	.0041	0	.0038	.0038	0	.077	.077
Material	0	.082	.099	0	.708	.85	0	1.80	2.16
Processing	0	.02	.01	0	.121	.06	0	.388	.194
Overhaul and Transportation	1.51	.60	.30	4.15	1.66	.83	17.0	6.80	3.40
TOTAL	1.51	1.19	.68	4.15	3.63	2.71	17.0	13.11	8.65

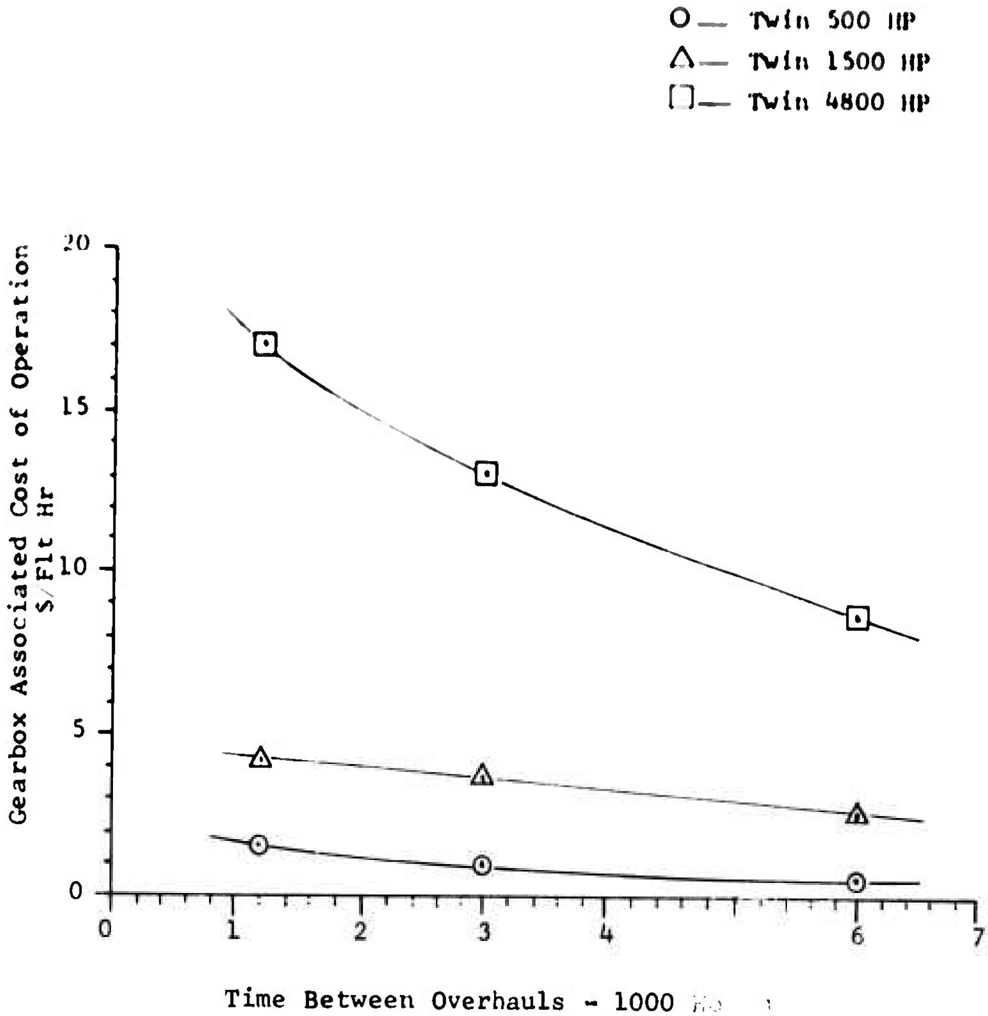

Figure 17. TBO VS Cost of Operati...

RECOMMENDATIONS

1. Design requirements for future Army helicopter transmission systems should specify extended TBO levels.

2. Maximum allowable surface compressive stress in low-speed gear teeth should be 160,000 psi for extended-life transmission design.

3. Gear and bearing designs should be optimized to provide the maximum ratio of EHD film thickness to composite surface roughness in load conjunctions to provide wear resistance for extended life.

4. Transmission development testing of several production prototype units, at a minimum of 125% overtorque for 50-100 hours duration, should be required prior to production to reveal design deficiencies which would preclude attainment of the objective extended life TBO.

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APPENDIX I
SPIRAL BEVEL GEAR DIMENSIONAL DATA

The dimensional data for all spiral bevel gears used in the nine study transmissions are tabulated in this Appendix. An optimization study was made for each gear application, based on speed, stress level, and bearing load level, with the resulting choice of gears shown in Tables XXII through XXXII.

TABLE XXII. INPUT SPIRAL BEVEL GEARS, TWIN 500 HP, 1200 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	6.000	-
Face Width, inches	1.200	1.200
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.243
Face Contact Ratio	-	1.763
Modified Contact Ratio	-	2.157
Outer Cone Distance, inches	-	6.893
Mean Cone Distance, inches	-	6.293
Circular Pitch, inches	0.524	-
Working Depth, inches	0.284	-
Whole Depth, inches	0.315	0.031
Clearance, inches	0.031	0.031
Pitch Diameter, inches	3.500	13.333
Addendum, inches	0.202	0.081
Dedendum, inches	0.113	0.234
Outside Diameter, inches	3.891	13.375
Pitch Apex to Crown, inches	6.615	1.671
Circular Thickness, inches	0.303	0.166
Mean Normal Top Land, inches	0.092	0.084
Outer Normal Top Land, inches	0.090	0.079
Inner Normal Top Land, inches	0.095	0.080
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.665	76.243
Root Angle, degrees	13.757	73.335
Dedendum Angle, degrees	0.951	1.957
Outer Spiral Angle, degrees	-	38.264
Mean Spiral Angle, degrees	-	35.000
Inner Spiral Angle, degrees	-	31.922
Hand of Spiral	LH	RH
Driving Member	PIN	-
Direction of Rotation	CW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	17.410

TABLE XXII - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	6.001	-
Cutter Radius, inches	6.000	-
Calc. Gear Finish Pt. Width, inches	-	0.102
Gear Finishing Point Width, inches	-	0.100
Roughing Point Width, inches	0.040	0.080
Outer Slot Width, inches	0.064	0.100
Mean Slot Width, inches	0.067	0.100
Inner Slot Width, inches	0.064	0.100
Finishing Cutter Blade Point, inches	0.040	0.065
Stock Allowance, inches	0.024	0.020
Max. Radius-Cutter Blades, inches	0.041	0.074
Max. Radius-Mutilation, inches	0.057	0.076
Max. Radius-Interference, inches	0.033	0.052
Cutter Edge Radius, inches	0.020	0.020
Calc. Cutter Number	3	6
Max. No. Blades in Cutter	22.351	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	2.908	-
Roughing Radial, inches	5.682	-
Geometry Factor-Strength-J	0.2808	0.2804
Strength Factor-Q	6.500	1.709
Factor	0.9047	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	0.001
Geometry Factor-Durability-I	0.1403	-
Durability Factor-Z	2786	1427
Geometry Factor-Scoring-G	0.003255	-
Scoring Factor-X	0.2586	-
Profile Sliding Factor	0.003341	0.005994
Root Line Face Width	1.200	1.200
Axial Factor-Driver CW	0.495 OUT	0.041 OUT
Axial Factor-Driver CCW	0.353 IN	0.100 OUT
Separating Factor-Driver CW	0.158 SEP	0.130 SEP
Separating Factor-Driver CCW	0.380 SEP	0.093 ATT

TABLE XXIII. TAIL ROTOR DRIVE SPIRAL BEVEL GEARS, TWIN 500 HP, 1200 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	6.034	-
Face Width, inches	0.600	0.600
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.246
Face Contact Ratio	-	0.889
Modified Contact Ratio	-	1.531
Outer Cone Distance, inches	-	6.854
Mean Cone Distance, inches	-	6.554
Circular Pitch, inches	0.521	-
Working Depth, inches	0.280	-
Whole Depth, inches	0.312	0.312
Clearance, inches	0.031	0.031
Pitch Diameter, inches	3.480	13.258
Addendum, inches	0.200	0.080
Dedendum, inches	0.111	0.232
Outside Diameter, inches	3.868	13.299
Pitch Apex to Crown, inches	6.578	1.663
Circular Thickness, inches	0.304	0.185
Mean Normal Top Land, inches	0.080	0.094
Outer Normal Top Land, inches	0.079	0.091
Inner Normal Top Land, inches	0.081	0.091
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.518	76.096
Root Angle, degrees	13.904	73.482
Dedendum Angle, degrees	0.804	1.810
Outer Spiral Angle, degrees	-	38.040
Mean Spiral Angle, degrees	-	36.390
Inner Spiral Angle, degrees	-	34.788
Hand of Spiral	LH	RH
Driving Member	GEAR	-
Direction of Rotation	CCW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	8.754

TABLE XXIII - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	6.001	-
Cutter Radius, inches	6.000	-
Calc. Gear Finish Pt. Width, inches	-	0.085
Gear Finishing Point Width, inches	-	0.090
Roughing Point Width, inches	0.050	0.070
Outer Slot Width, inches	0.076	0.090
Mean Slot Width, inches	0.076	0.090
Inner Slot Width, inches	0.076	0.090
Finishing Cutter Blade Point, inches	0.040	0.050
Stock Allowance, inches	0.026	0.020
Max. Radius-Cutter Blades, inches	0.046	0.060
Max. Radius-Mutilation, inches	0.078	0.085
Max. Radius-Interference, inches	0.031	0.052
Cutter Edge Radius, inches	0.020	0.020
Calc. Cutter Number	2	6
Max. No. Blades in Cutter	38.735	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	2.614	-
Roughing Radial, inches	5.583	-
Geometry Factor-Strength-J	0.2114	0.2113
Strength Factor-Q	17.445	4.580
Factor	1.3065	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	-
Geometry Factor-Durability-I	0.1313	-
Durability Factor-Z	4095	2098
Geometry Factor-Scoring-G	0.003965	-
Scoring Factor-X	0.4152	-
Profile Sliding Factor	0.003471	0.006276
Root Line Face Width	0.600	0.600
Axial Factor-Driver CW	0.497 OUT	0.039 OUT
Axial Factor-Driver CCW	0.359 IN	0.099 OUT
Separating Factor-Driver CW	0.150 SEP	0.131 SEP
Separating Factor-Driver CCW	0.375 SEP	0.094 ATT

TABLE XXIV. OIL PUMP DRIVE SPIRAL BEVEL GEARS, TWIN 500 HP, 1200 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	6.774	-
Face Width, inches	0.280	0.280
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.275
Face Contact Ratio	-	0.406
Modified Contact Ratio	-	1.338
Outer Cone Distance, inches	-	6.105
Mean Cone Distance, inches	-	5.965
Circular Pitch, inches	0.464	-
Working Depth, inches	0.252	-
Whole Depth, inches	0.280	0.280
Clearance, inches	0.028	0.028
Pitch Diameter, inches	3.100	11.810
Addendum, inches	0.180	0.072
Dedendum, inches	0.100	0.207
Outside Diameter, inches	3.447	11.847
Pitch Apex to Crown, inches	5.859	1.480
Circular Thickness, inches	0.263	0.181
Mean Normal Top Land, inches	0.068	0.101
Outer Normal Top Land, inches	0.067	0.099
Inner Normal Top Land, inches	0.069	0.099
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.836	76.414
Root Angle, degrees	13.586	73.164
Dedendum Angle, degrees	1.122	2.128
Outer Spiral Angle, degrees	-	34.011
Mean Spiral Angle, degrees	-	33.290
Inner Spiral Angle, degrees	-	32.578
Hand of Spiral	LH	RH
Driving Member	GEAR	-
Direction of Rotation	CCW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	4.586

TABLE XXIV - Continued		
	Pinion	Gear
Theoretical Cutter Radius, inches	6.002	-
Cutter Radius, inches	6.000	-
Calc. Gear Finish Pt. Width, inches	-	0.082
Gear Finishing Point Width, inches	-	0.080
Roughing Point Width, inches	0.060	0.060
Outer Slot Width, inches	0.086	0.080
Mean Slot Width, inches	0.086	0.080
Inner Slot Width, inches	0.086	0.080
Finishing Cutter Blade Point, inches	0.050	0.050
Stock Allowance, inches	0.026	0.020
Max. Radius-Cutter Blades, inches	0.060	0.060
Max. Radius-Mutilation, inches	0.078	0.068
Max. Radius-Interference, inches	0.026	0.047
Cutter Edge Radius, inches	0.015	0.015
Calc. Cutter Number	3	7
Max. No. Blades in Cutter	72.814	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	3.250	-
Roughing Radial, inches	5.683	-
Geometry Factor-Strength-J	0.1825	0.1819
Strength Factor-Q	53.019	13.961
Factor	1.4948	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	0.002
Geometry Factor-Durability-I	0.1069	-
Durability Factor-Z	7458	3821
Geometry Factor-Scoring-G	0.004888	-
Scoring Factor-X	0.7418	-
Profile Sliding Factor	0.003233	0.005942
Root Line Face Width	0.280	0.280
Axial Factor-Driver CW	0.492 OUT	0.044 OUT
Axial Factor-Driver CCW	0.346 IN	0.102 OUT
Separating Factor-Driver CW	0.168 SEP	0.129 SEP
Separating Factor-Driver CCW	0.388 SEP	0.091 ATT

TABLE XXV. INPUT SPIRAL BEVEL GEARS, TWIN 500 HP, 3000 AND 6000 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	5.500	-
Face Width, inches	1.400	1.400
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.241
Face Contact Ratio	-	1.899
Modified Contact Ratio	-	2.269
Outer Cone Distance, inches	-	7.519
Mean Cone Distance, inches	-	6.819
Circular Pitch, inches	0.571	-
Working Depth, inches	0.312	-
Whole Depth, inches	0.346	0.346
Clearance, inches	0.034	0.034
Pitch Diameter, inches	3.818	14.545
Addendum, inches	0.222	0.090
Dedendum, inches	0.124	0.256
Outside Diameter, inches	4.248	14.591
Pitch Apex to Crown, inches	7.216	1.822
Circular Thickness, inches	0.331	0.178
Mean Normal Top Land, inches	0.102	0.090
Outer Normal Top Land, inches	0.099	0.085
Inner Normal Top Land, inches	0.107	0.085
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.761	76.339
Root Angle, degrees	13.661	73.238
Dedendum Angle, degrees	1.048	2.053
Outer Spiral Angle, degrees	-	38.194
Mean Spiral Angle, degrees	-	35.000
Inner Spiral Angle, degrees	-	32.030
Hand of Spiral	LH	RH
Driving Member	PIN	-
Direction of Rotation	CW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	18.619

TABLE XXV - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	6.802	-
Cutter Radius, inches	6.800	-
Calc. Gear Finish Pt. Width, inches	-	0.111
Gear Finishing Point Width, inches	-	0.110
Roughing Point Width, inches	0.040	0.090
Outer Slot Width, inches	0.067	0.110
Mean Slot Width, inches	0.070	0.110
Inner Slot Width, inches	0.067	0.110
Finishing Cutter Blade Point, inches	0.040	0.065
Stock Allowance, inches	0.027	0.020
Max. Radius-Cutter Blades, inches	0.044	0.074
Max. Radius-Mutilation, inches	0.063	0.093
Max. Radius-Interference, inches	0.037	0.058
Cutter Edge Radius, inches	0.020	0.020
Calc. Cutter Number	3	7
Max. No. Blades in Cutter	21.901	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	3.101	-
Roughing Radial, inches	6.289	-
Geometry Factor-Strength-J	0.2881	0.2872
Strength Factor-Q	4.664	1.228
Factor	0.8262 MN	
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	0.002
Geometry Factor-Durability-I	0.1441	-
Durability Factor-Z	2332	1195
Geometry Factor-Scoring-G	0.003255	-
Scoring Factor-X	0.2357	-
Profile Sliding Factor	0.003617	0.006476
Root Line Face Width	1.400	1.400
Axial Factor-Driver CW	0.456 OUT	0.038 OUT
Axial Factor-Driver CCW	0.326 IN	0.092 OUT
Separating Factor-Driver CW	0.146 SEP	0.120 SEP
Separating Factor-Driver CCW	0.031 SEP	0.086 ATT

TABLE XXVI. TAIL ROTOR DRIVE SPIRAL BEVEL GEARS, TWIN 500 HP, 3000 AND 6000 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	5.530	-
Face Width, inches	0.700	0.700
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.253
Face Contact Ratio	-	0.918
Modified Contact Ratio	-	1.553
Outer Cone Distance, inches	-	7.478
Mean Cone Distance, inches	-	7.128
Circular Pitch, inches	0.568	-
Working Depth, inches	0.307	-
Whole Depth, inches	0.341	0.341
Clearance, inches	0.034	0.034
Pitch Diameter, inches	3.797	14.467
Addendum, inches	0.219	0.088
Dedendum, inches	0.122	0.253
Outside Diameter, inches	4.222	14.511
Pitch Apex to Crown, inches	7.178	1.814
Circular Thickness, inches	0.342	0.191
Mean Normal Top Land, inches	0.100	0.096
Outer Normal Top Land, inches	0.099	0.093
Inner Normal Top Land, inches	0.102	0.093
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.653	76.231
Root Angle, degrees	13.769	73.347
Dedendum Angle, degrees	0.939	1.945
Outer Spiral Angle, degrees	-	37.006
Mean Spiral Angle, degrees	-	35.360
Inner Spiral Angle, degrees	-	33.765
Hand of Spiral	LH	RH
Driving Member	GEAR	-
Direction of Rotation	CCW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	9.360

TABLE XXVI - Continued		
	Pinion	Gear
Theoretical Cutter Radius, inches	6.801	-
Cutter Radius, inches	6.800	-
Calc. Gear Finish Pt. Width, inches	-	0.110
Gear Finishing Point Width, inches	-	0.110
Roughing Point Width, inches	0.050	0.090
Outer Slot Width, inches	0.076	0.110
Mean Slot Width, inches	0.076	0.110
Inner Slot Width, inches	0.075	0.110
Finishing Cutter Blade Point, inches	0.040	0.065
Stock Allowance, inches	0.025	0.020
Max. Radius-Cutter Blades, inches	0.046	0.074
Max. Radius-Mutilation, inches	0.077	0.093
Max. Radius-Interference, inches	0.035	0.059
Cutter Edge Radius, inches	0.003	0.003
Calc. Cutter Number	3	6
Max. No. Blades in Cutter	38.642	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	2.884	-
Roughing Radial, inches	6.399	-
Geometry Factor-Strength-J	0.1951	0.1588
Strength Factor-Q	13.908	4.486
Factor	1.2875	-
Strength Balance Desired	GIVN	-
Strength Balance Obtained	GIVN	0.154
Geometry Factor-Durability-I	0.1306	-
Durability Factor-Z	3484	1785
Geometry Factor-Scoring-G	0.000025	-
Scoring Factor-X	0.3672	-
Profile Sliding Factor	0.003800	0.006890
Root Line Face Width	0.700	0.700
Axial Factor-Driver CW	0.442 OUT	0.036 OUT
Axial Factor-Driver CCW	0.317 IN	0.089 OUT
Separating Factor-Driver CW	0.139 SEP	0.116 SEP
Separating Factor-Driver CCW	0.338 SEP	0.083 ATT

TABLE XXVII. OIL PUMP DRIVE SPIRAL BEVEL GEARS, TWIN 500 HP, 3000 AND 6000 TBO

	Pinion	Gear
Number of Teeth	21	80
Diametral Pitch	6.360	-
Face Width, inches	0.350	0.350
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.283
Face Contact Ratio	-	0.456
Modified Contact Ratio	-	1.361
Outer Cone Distance, inches	-	6.502
Mean Cone Distance, inches	-	6.327
Circular Pitch, inches	0.494	-
Working Depth, inches	0.269	-
Whole Depth, inches	0.299	0.299
Clearance, inches	0.030	0.030
Pitch Diameter, inches	3.302	12.579
Addendum, inches	0.192	0.078
Dedendum, inches	0.107	0.221
Outside Diameter, inches	3.673	12.618
Pitch Apex to Crown, inches	6.241	1.576
Circular Thickness, inches	0.305	0.167
Mean Normal Top Land, inches	0.099	0.088
Outer Normal Top Land, inches	0.098	0.085
Inner Normal Top Land, inches	0.100	0.085
Pitch Angle, degrees	14.708	75.292
Face Angle of Blank, degrees	16.997	76.575
Root Angle, degrees	13.425	73.002
Dedendum Angle, degrees	1.284	2.289
Outer Spiral Angle, degrees	-	32.774
Mean Spiral Angle, degrees	-	32.020
Inner Spiral Angle, degrees	-	31.278
Hand of Spiral	LH	RH
Driving Member	GEAR	-
Direction of Rotation	CCW	-
Backlash, inches	0.005 MIN	0.007 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	5.383

TABLE XXVII - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	6.803	-
Cutter Radius, inches	6.800	-
Calc. Gear Finish Pt. Width, inches	-	0.110
Gear Finishing Point Width, inches	-	0.110
Roughing Point Width, inches	0.050	0.090
Outer Slot Width, inches	0.071	0.110
Mean Slot Width, inches	0.072	0.110
Inner Slot Width, inches	0.071	0.110
Finishing Cutter Blade Point, inches	0.040	0.065
Stock Allowance, inches	0.021	0.020
Max. Radius-Cutter Blades, inches	0.046	0.074
Max. Radius-Mutilation, inches	0.070	0.093
Max. Radius-Interference, inches	0.029	0.051
Cutter Edge Radius, inches	0.003	0.003
Calc. Cutter Number	4	7
Max. No. Blades in Cutter	69.677	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	3.573	-
Roughing Radial, inches	6.376	-
Geometry Factor-Strength-J	0.1836	0.1385
Strength Factor-Q	37.760	13.135
Factor	1.4694	-
Strength Balance Desired	GIVN	-
Strength Balance Obtained	GIVN	0.210
Geometry Factor-Durability-I	0.1077	-
Durability Factor-Z	6239	3196
Geometry Factor-Scoring-G	0.000066	-
Scoring Factor-X	0.6469	-
Profile Sliding Factor	0.003453	0.006369
Root Line Face Width	0.350	0.350
Axial Factor-Driver CW	0.444 OUT	0.042 OUT
Axial Factor-Driver CCW	0.309 IN	0.094 OUT
Separating Factor-Driver CW	0.160 SEP	0.177 SEP
Separating Factor-Driver CCW	0.357 SEP	0.081 ATTN

TABLE XXVIII. INPUT SPIRAL BEVEL GEARS, TWIN 1500 HP; 1200, 3000 AND 6000 TBO

	Pinion	Gear
Number of Teeth	46	93
Diametral Pitch	10.000	-
Face Width, inches	1.200	1.200
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	85.000	-
Transverse Contact Ratio	-	1.406
Face Contact Ratio	-	2.027
Modified Contact Ratio	-	2.467
Outer Cone Distance, inches	-	5.385
Mean Cone Distance, inches	-	4.785
Circular Pitch, inches	0.314	-
Working Depth, inches	0.171	-
Whole Depth, inches	0.190	0.190
Clearance, inches	0.019	0.019
Pitch Diameter, inches	4.600	9.300
Addendum, inches	0.114	0.057
Dedendum, inches	0.076	0.133
Outside Diameter, inches	4.806	9.358
Pitch Apex to Crown, inches	4.820	2.666
Circular Thickness, inches	0.165	0.108
Mean Normal Top Land, inches	0.072	0.062
Outer Normal Top Land, inches	0.070	0.056
Inner Normal Top Land, inches	0.073	0.057
Pitch Angle, degrees	25.285	59.715
Face Angle of Blank, degrees	26.753	60.582
Root Angle, degrees	24.418	58.247
Dedendum Angle, degrees	0.867	1.468
Outer Spiral Angle, degrees	-	31.777
Mean Spiral Angle, degrees	-	25.000
Inner Spiral Angle, degrees	-	18.164
Hand of Spiral	LH	RH
Driving Member	PIN	-
Direction of Rotation	CW	-
Backlash, inches	0.004 MIN	0.006 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	22.285

TABLE XXVIII - Continued		
	Pinion	Gear
Theoretical Cutter Radius, inches	3.751	-
Cutter Radius, inches	3.750	-
Calc. Gear Finish Pt. Width, inches	-	0.074
Gear Finishing Point Width, inches	-	0.070
Roughing Point Width, inches	0.040	0.060
Outer Slot Width, inches	0.049	0.070
Mean Slot Width, inches	0.053	0.070
Inner Slot Width, inches	0.049	0.070
Finishing Cutter Blade Point, inches	0.030	0.040
Stock Allowance, inches	0.009	0.010
Max. Radius-Cutter Blades, inches	0.031	0.046
Max. Radius-Mutilation, inches	0.049	0.068
Max. Radius-Interference, inches	0.018	0.023
Cutter Edge Radius, inches	0.010	0.020
Calc. Cutter Number	2	3
Max. No. Blades in Cutter	-	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	2.334	-
Roughing Radial, inches	4.668	-
Geometry Factor-Strength-J	0.3630	0.3614
Strength Factor-Q	5.613	2.788
Factor	0.7138 MN	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	0.006
Geometry Factor-Durability-I	0.1112	-
Durability Factor-Z	2381	1674
Geometry Factor-Scoring-G	0.000974	-
Scoring Factor-X	0.1141	-
Profile Sliding Factor	0.002384	0.002594
Root Line Face Width	1.200	1.200
Axial Factor-Driver CW	0.290 OUT	0.027 OUT
Axial Factor-Driver CCW	0.122 IN	0.141 OUT
Separating Factor-Driver CW	0.080 SEP	0.146 SEP
Separating Factor-Driver CCW	0.275 SEP	0.048 ATT

TABLE XXIX. TAIL ROTOR DRIVE SPIRAL BEVEL GEAR, TWIN 1500 HP; 1200, 3000, AND 6000 TBO

	Pinion	Gear
Number of Teeth	35	37
Diametral Pitch	7.000	-
Face Width, inches	1.200	1.200
Pressure Angle, degrees	20.000	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.338
Face Contact Ratio	-	1.888
Modified Contact Ratio	-	2.314
Outer Cone Distance, inches	-	3.638
Mean Cone Distance, inches	-	3.038
Circular Pitch, inches	0.449	-
Working Depth, inches	0.258	-
Whole Depth, inches	0.284	0.284
Clearance, inches	0.027	0.027
Pitch Diameter, inches	5.000	5.286
Addendum, inches	0.135	0.123
Dedendum, inches	0.150	0.161
Outside Diameter, inches	5.196	5.455
Pitch Apex to Crown, inches	2.550	2.411
Circular Thickness, inches	0.182	0.186
Mean Normal Top Land, inches	0.079	0.091
Outer Normal Top Land, inches	0.076	0.083
Inner Normal Top Land, inches	0.083	0.084
Pitch Angle, degrees	43.409	46.591
Face Angle of Blank, degrees	46.535	49.533
Root Angle, degrees	40.467	43.465
Dedendum Angle, degrees	2.942	3.126
Outer Spiral Angle, degrees	-	35.088
Mean Spiral Angle, degrees	-	30.000
Inner Spiral Angle, degrees	-	25.501
Hand of Spiral	RH	LH
Driving Member	PIN	-
Direction of Rotation	CCW	-
Backlash, inches	0.004 MIN	0.006 MAX
Tooth Taper	TRLM	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	32.986

TABLE XXIX - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	3.315	-
Cutter Radius, inches	3.500	-
Calc. Gear Finish Pt. Width, inches	-	0.075
Gear Finishing Point Width, inches	-	0.070
Roughing Point Width, inches	0.060	0.060
Outer Slot Width, inches	0.075	0.070
Mean Slot Width, inches	0.078	0.070
Inner Slot Width, inches	0.071	0.070
Finishing Cutter Blade Point, inches	0.040	0.040
Stock Allowance, inches	0.011	0.010
Max. Radius-Cutter Blades, inches	0.046	0.046
Max. Radius-Mutilation, inches	0.069	0.068
Max. Radius-Interference, inches	0.033	0.034
Cutter Edge Radius, inches	0.015	0.015
Calc. Cutter Number	8	9
Max. No. Blades in Cutter	-	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	6.346	-
Roughing Radial, inches	3.293	-
Geometry Factor-Strength-J	0.2611	0.2602
Strength Factor-O	5.494	5.215
Factor	0.7972 MN	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	STRS	0.063
Geometry Factor-Durability-I	0.0762	-
Durability Factor-Z	2645	257[illegible]
Geometry Factor-Scoring-G	0.000977	-
Scoring Factor-X	0.0900	-
Profile Sliding Factor	0.002899	0.002811
Root Line Face Width	1.200	1.200
Axial Factor-Driver CW	0.063 IN	0.318 OUT
Axial Factor-Driver CCW	0.339 OUT	0.041 IN
Separating Factor-Driver CW	0.336 SEP	0.059 ATT
Separating Factor-Driver CCW	0.044 ATT	0.321 SEP

TABLE XXX. ENGINE GEARBOX SPIRAL BEVEL GEARS, TWIN 4800 HP; 1200, 3000, AND 6000 TBO

	Pinion	Gear
Number of Teeth	67	99
Diametral Pitch	7.750	-
Face Width, inches	2.500	2.500
Pressure Angle, degrees	22.500	-
Shaft Angle, degrees	60.000	-
Transverse Contact Ratio	-	1.358
Face Contact Ratio	-	2.576
Modified Contact Ratio	-	2.912
Outer Cone Distance, inches	-	10.776
Mean Cone Distance, inches	-	9.526
Circular Pitch, inches	0.405	-
Working Depth, inches	0.216	-
Whole Depth, inches	0.240	0.240
Clearance, inches	0.024	0.024
Pitch Diameter, inches	8.645	12.774
Addendum, inches	0.128	0.088
Dedendum, inches	0.122	0.153
Outside Diameter, inches	8.880	12.915
Pitch Apex to Crown, inches	9.819	8.627
Circular Thickness, inches	0.192	0.153
Mean Normal Top Land, inches	0.084	0.080
Outer Normal Top Land, inches	0.083	0.066
Inner Normal Top Land, inches	0.085	0.068
Pitch Angle, degrees	23.649	36.351
Face Angle of Blank, degrees	24.390	36.875
Root Angle, degrees	23.125	35.610
Dedendum Angle, degrees	0.524	0.741
Outer Spiral Angle, degrees	-	29.907
Mean Spiral Angle, degrees	-	20.000
Inner Spiral Angle, degrees	-	9.765
Hand of Spiral	LH	RH
Driving Member	PIN	-
Direction of Rotation	CW	-
Backlash, inches	0.007 MIN	0.012 MAX
Tooth Taper	TRLM	
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	23.200

TABLE XXX - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	6.000	-
Cutter Radius, inches	6.000	-
Calc. Gear Finish Pt. Width, inches	-	0.085
Gear Finishing Point Width, inches	-	0.080
Roughing Point Width, inches	0.035	0.060
Outer Slot Width, inches	0.060	0.080
Mean Slot Width, inches	0.068	0.080
Inner Slot Width, inches	0.060	0.080
Finishing Cutter Blade Point, inches	0.035	0.050
Stock Allowance, inches	0.025	0.020
Max. Radius-Cutter Blades, inches	0.037	0.063
Max. Radius-Mutilation, inches	0.063	0.072
Max. Radius-Interference, inches	0.020	0.023
Cutter Edge Radius, inches	0.010	0.020
Calc. Cutter Number	1	1
Max. No. Blades in Cutter	-	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	1.265	-
Roughing Radial, inches	9.362	-
Geometry Factor-Strength-J	0.4079	0.4329
Strength Factor-O	1.054	0.672
Factor	0.5657 MN	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	PW	0.152
Geometry Factor-Durability-I	0.0880	-
Durability Factor-Z	986	812
Geometry Factor-Scoring-G	0.000485	-
Scoring Factor-X	0.0418	-
Profile Sliding Factor	0.003334	0.003462
Root Line Face Width	2.500	2.500
Axial Factor-Driver CW	0.134 OUT	0.006 IN
Axial Factor-Driver CCW	0.041 IN	0.098 OUT
Separating Factor-Driver CW	0.067 SEP	0.101 SEP
Separating Factor-Driver CCW	0.144 SEP	0.025 SEP

TABLE XXXI. INPUT SPIRAL BEVEL GEARS, 4800 HP; 1200, 3000, AND 6000 TBO

	Pinion	Gear
Number of Teeth	86	231
Diametral Pitch	7.750	-
Face Width, inches	2.750	2.750
Pressure Angle, degrees	22.500	-
Shaft Angle, degrees	85.000	-
Transverse Contact Ratio	-	1.999
Face Contact Ratio	-	2.441
Modified Contact Ratio	-	16.412
Outer Cone Distance, inches	-	15.037
Mean Cone Distance, inches	-	15.037
Circular Pitch, inches	0.405	-
Working Depth, inches	0.221	-
Whole Depth, inches	0.245	0.245
Clearance, inches	0.024	0.024
Pitch Diameter, inches	11.097	29.806
Addendum, inches	0.152	0.069
Dedendum, inches	0.093	0.177
Outside Diameter, inches	11.384	29.864
Pitch Apex to Crown, inches	15.394	6.811
Circular Thickness, inches	0.229	0.130
Mean Normal Top Land, inches	0.103	0.075
Outer Normal Top Land, inches	0.102	0.066
Inner Normal Top Land, inches	0.104	0.067
Pitch Angle, degrees	19.759	65.241
Face Angle of Blank, degrees	20.408	65.596
Root Angle, degrees	19.404	64.592
Dedendum Angle, degrees	0.355	0.648
Outer Spiral Angle, degrees	-	22.551
Mean Spiral Angle, degrees	-	15.000
Inner Spiral Angle, degrees	-	7.146
Hand of Spiral	LH	RH
Driving Member	PIN	-
Direction of Rotation	CW	-
Backlash, inches	0.007 MIN	0.012 MAX
Tooth Taper	TRIM	-
Cutting Method	-	SS
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	16.756

TABLE XXXI - Continued

	Pinion	Gear
Theoretical Cutter Radius, inches	9.000	-
Cutter Radius, inches	9.000	-
Calc. Gear Finish Pt. Width, inches	-	0.104
Gear Finishing Point Width, inches	-	0.100
Roughing Point Width, inches	0.050	0.090
Outer Slot Width, inches	0.059	0.100
Mean Slot Width, inches	0.063	0.100
Inner Slot Width, inches	0.058	0.100
Finishing Cutter Blade Point, inches	0.040	0.065
Stock Allowance, inches	0.008	0.010
Max. Radius-Cutter Blades, inches	0.039	0.070
Max. Radius-Mutilation, inches	0.050	0.081
Max. Radius-Interference, inches	0.018	0.023
Cutter Edge Radius, inches	0.010	0.010
Calc. Cutter Number	0	1
Max. No. Blades in Cutter	19.259	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	1.004	-
Roughing Radial, inches	15.397	-
Geometry Factor-Strength-J	0.4311	0.4398
Strength Factor-Q	0.706	0.258
Factor	0.7263 MN	-
Strength Balance Desired	STRS	-
Strength Balance Obtained	TPLD	0.020
Geometry Factor-Durability-I	0.1195	-
Durability Factor-Z	629	384
Geometry Factor-Scoring-G	0.000377	-
Scoring Factor-X	0.0335	-
Profile Sliding Factor	0.003128	0.003893
Root Line Face Width	2.750	2.750
Axial Factor-Driver CW	0.078 OUT	0.020 OUT
Axial Factor-Driver CCW	0.021 IN	0.037 OUT
Separating Factor-Driver CW	0.062 SEP	0.031 SEP
Separating Factor-Driver CCW	0.097 SEP	0.005 ATT

TABLE XXXII. TAIL ROTOR DRIVE SPIRAL BEVEL GEARS, 4800 HP; 1200, 3000, AND 6000 TBO

	Pinion	Gear
Number of Teeth	106	231
Diametral Pitch	8.856	-
Face Width, inches	0.700	0.700
Pressure Angle, degrees	22.500	-
Shaft Angle, degrees	90.000	-
Transverse Contact Ratio	-	1.702
Face Contact Ratio	-	0.327
Modified Contact Ratio	-	1.733
Outer Cone Distance, inches	-	14.350
Mean Cone Distance, inches	-	14.000
Circular Pitch, inches	0.355	-
Working Depth, inches	0.226	-
Whole Depth, inches	0.249	0.249
Clearance, inches	0.023	0.023
Pitch Diameter, inches	11.969	26.084
Addendum, inches	0.154	0.072
Dedendum, inches	0.095	0.0177
Outside Diameter, inches	12.249	26.144
Pitch Apex to Crown, inches	12.978	5.919
Circular Thickness, inches	0.233	0.100
Mean Normal Top Land, inches	0.101	0.041
Outer Normal Top Land, inches	0.101	0.036
Inner Normal Top Land, inches	0.101	0.034
Pitch Angle, degrees	24.649	65.351
Face Angle of Blank, degrees	25.356	65.731
Root Angle, degrees	24.269	64.643
Dedendum Angle, degrees	0.380	0.707
Outer Spiral Angle, degrees	-	11.200
Mean Spiral Angle, degrees	-	9.190
Inner Spiral Angle, degrees	-	7.147
Hand of Spiral	LH	RH
Driving Member	GEAR	-
Direction of Rotation	CCW	-
Backlash, inches	0.007 MIN	0.012 MIN
Tooth Taper	-	-
Cutting Method	-	SB
Gear Type	-	GENERATED
Face Width in Percent of Cone Distance	-	4.878

TABLE XXXII - Continued		
	Pinion	Gear
Theoretical Cutter Radius, inches	9.000	-
Cutter Radius, inches	9.000	-
Calc. Gear Finish Pt. Width, inches	-	0.100
Gear Finishing Point Width, inches	-	0.100
Roughing Point Width, inches	0.025	0.090
Outer Slot Width, inches	0.030	0.100
Mean Slot Width, inches	0.029	0.100
Inner Slot Width, inches	0.028	0.100
Finishing Cutter Blade Point, inches	0.020	0.065
Stock Allowance, inches	0.003	0.010
Max. Radius-Cutter Blades, inches	0.012	0.070
Max. Radius-Mutilation, inches	0.029	0.081
Max. Radius-Interference, inches	0.015	0.021
Cutter Edge Radius, inches	0.010	0.010
Calc. Cutter Number	0	0
Max. No. Blades in Cutter	74.577	-
Cutter Blades Required	STD. DEPTH	STD. DEPTH
Duplex Sum of Dedendum Angle, degrees	-	-
Roughing Radial, inches	15.386	-
Geometry Factor-Strength-J	0.4857	0.3000
Strength Factor-O	2.523	1.875
Factor	1.1540	-
Strength Balance Desired	GIVN	-
Strength Balance Obtained	GIVN	0.619
Geometry Factor-Durability-I	0.1340	-
Durability Factor-Z	1091	739
Geometry Factor-Scoring-G	0.00034	-
Scoring Factor-X	0.0599	-
Profile Sliding Factor	0.003499	0.003738
Root Line Face Width	0.700	0.700
Axial Factor-Driver CW	0.055 OUT	0.025 OUT
Axial Factor-Driver CCW	0.005 OUT	0.035 OUT
Separating Factor-Driver CW	0.054 SEP	0.025 SEP
Separating Factor-Driver CCW	0.077 SEP	0.002 SEP

APPENDIX II
FUNCTIONAL GEAR DATA

The functional spur gear data shown in Tables II, IV, VII, IX, XII, and XIV are shown in graphic form in Figures 18 through 57. Each spur gear was analyzed at 100% transmitted load, and the resulting operational characteristics are shown graphically as a function of driving gear angle of roll from first point of contact to last point of contact. Effects of gear profile modification and predicted gear tooth deflections are represented in the charts.

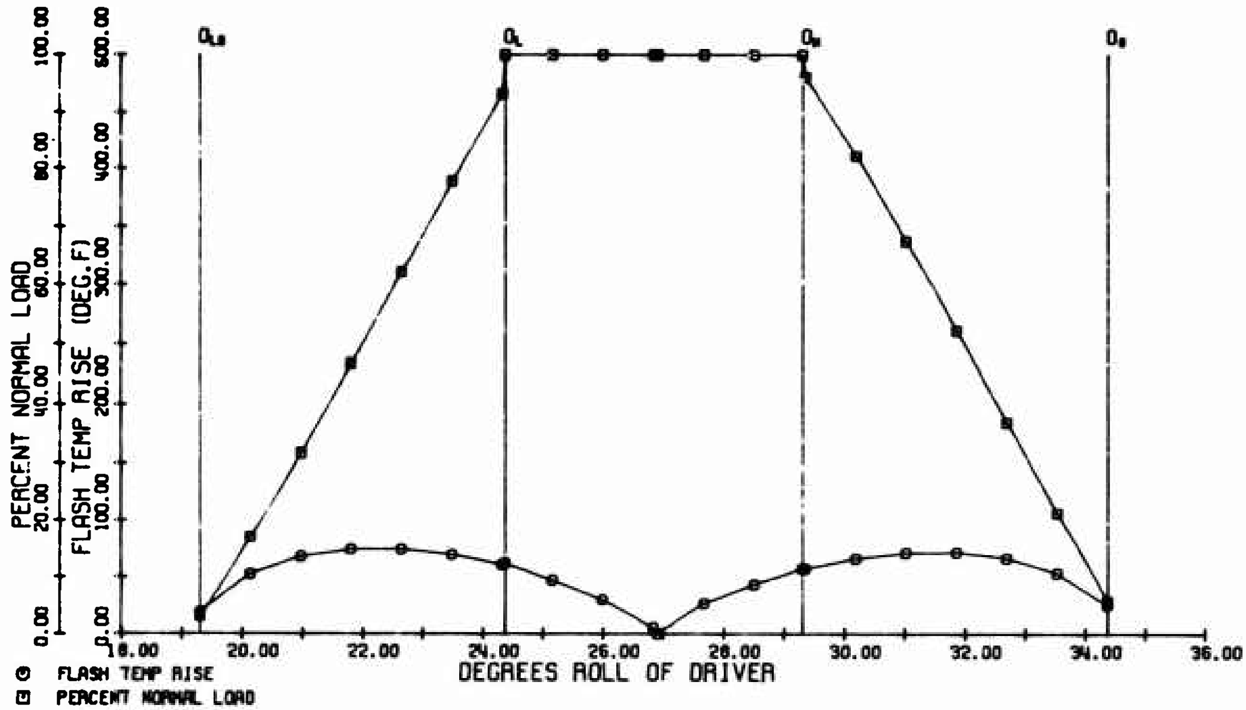

Figure 18. Flash Temperature Rise and Load Diagram, Twin 500 HP; 1200-Hour TBO, Sun-Planet Mesh.

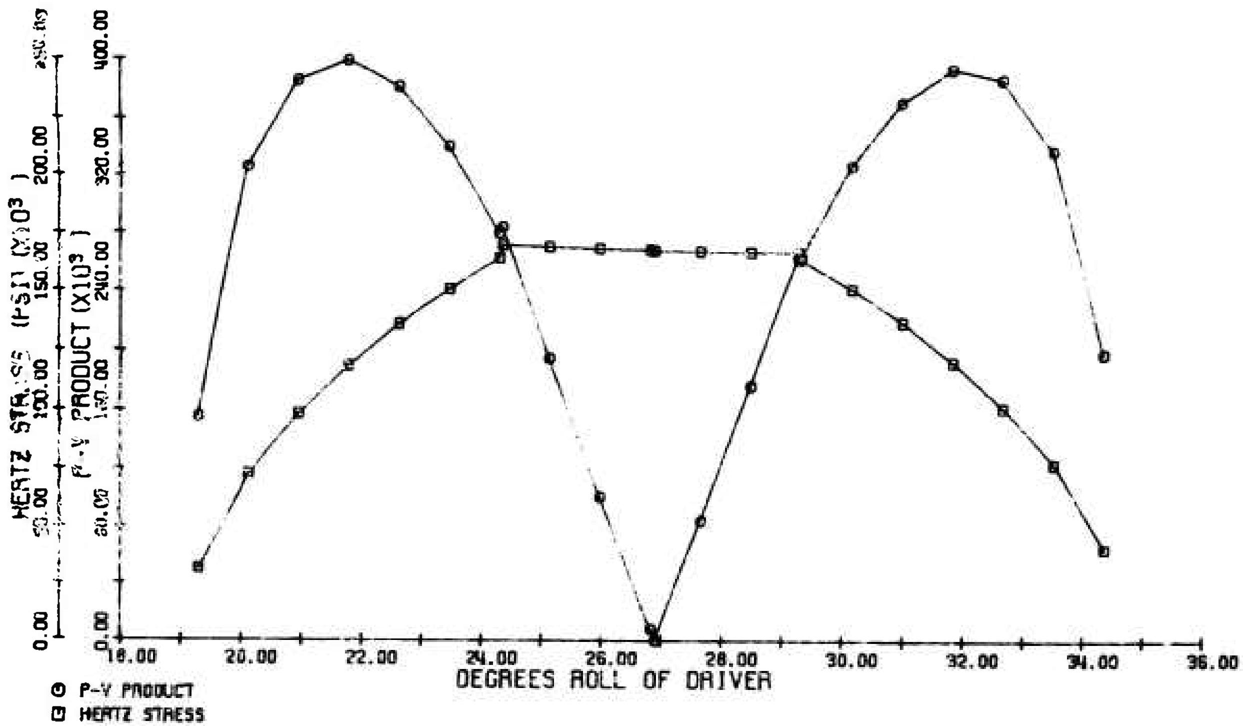

Figure 19. Hertz Stress and P-V Product Diagram, Twin 500 HP, 1200-Hour TBO, Sun-Planet Mesh.

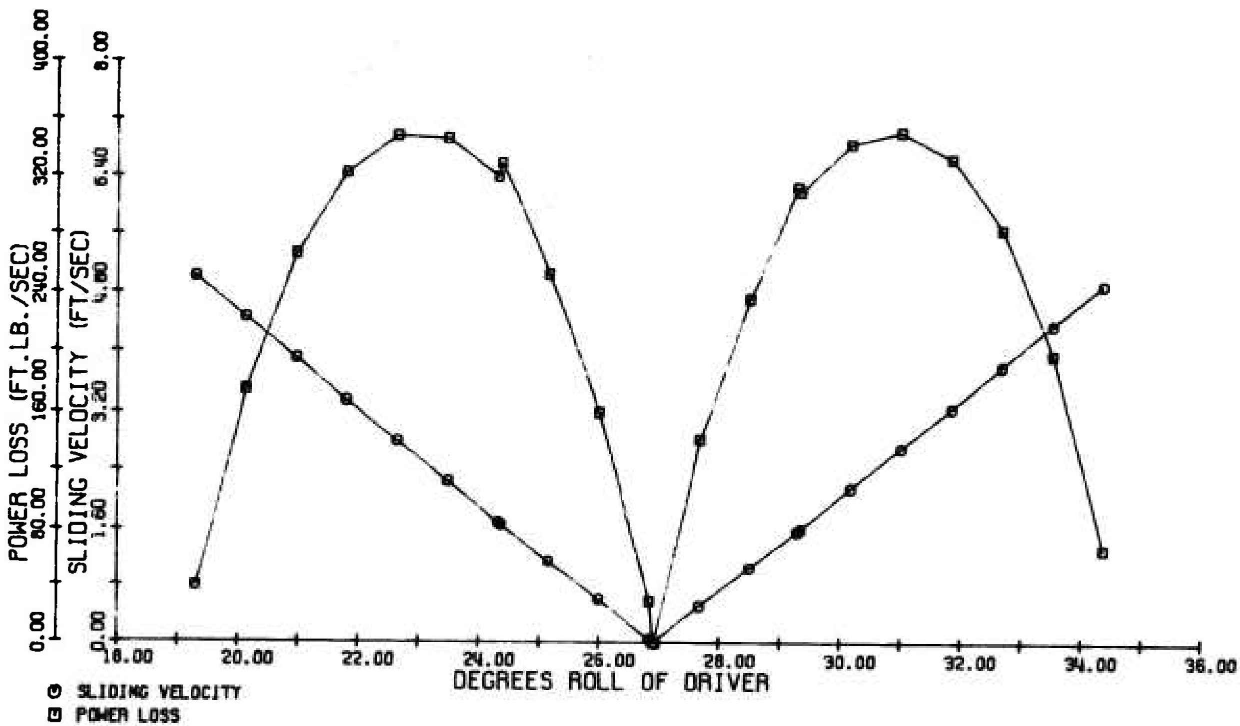

Figure 20. Sliding Velocity and Power Loss Diagram, Twin 500 HP, 1200-Hour TBO, Sun-Planet Mesh.

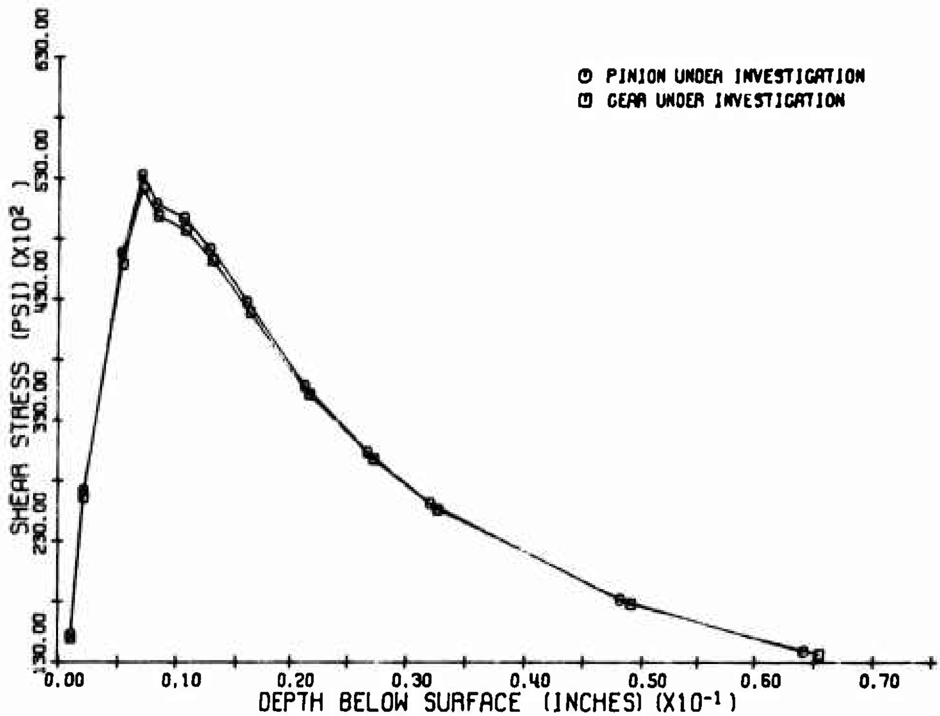

Figure 21. Shear Stress Diagram, Twin 500 HP, 1200-Hour TBO, Sun-Planet Mesh.

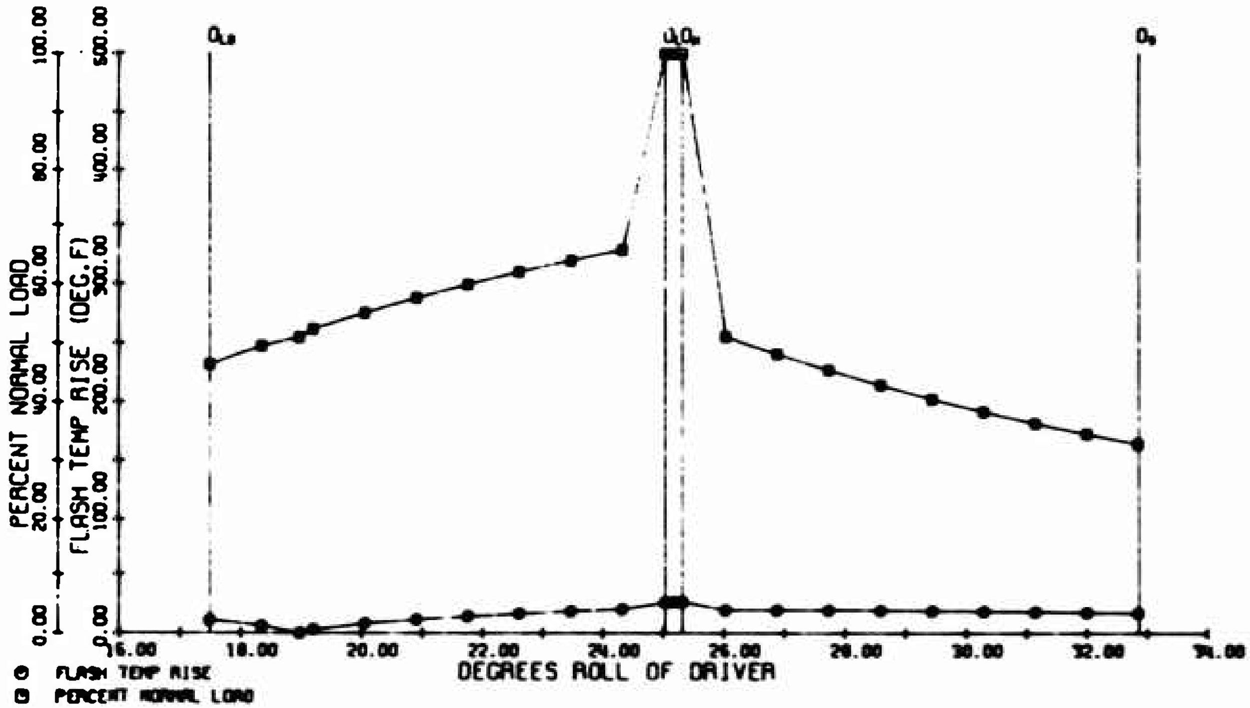

Figure 22. Flash Temperature Rise and Load Diagram, Twin 500 HP, 1200-Hour TBO, Ring-Planet Mesh.

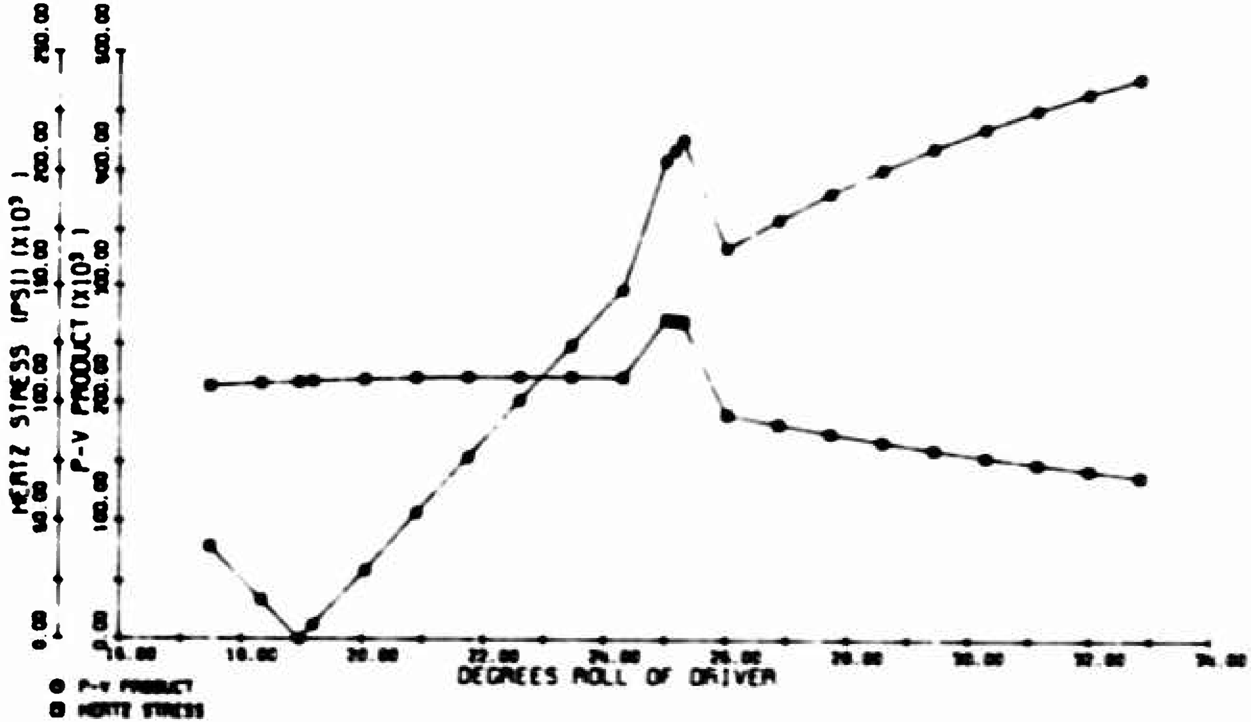

Figure 23. Hertz Stress and P-V Product Diagram, Twin 500 HP, 1200-Hour TBO, Ring-Planet Mesh.

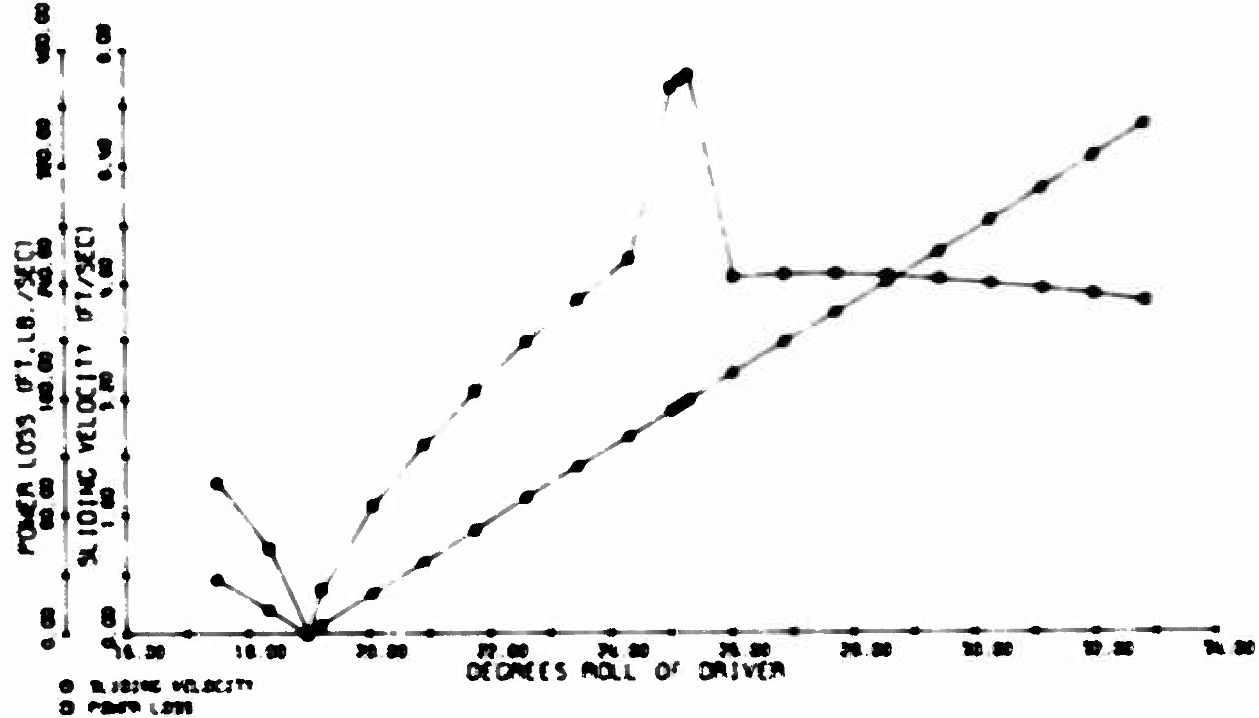

Figure 24. Sliding Velocity and Power Loss Diagram. Twin 500 HP, 1200-Hour TBO, Ring-Planet Mesh.

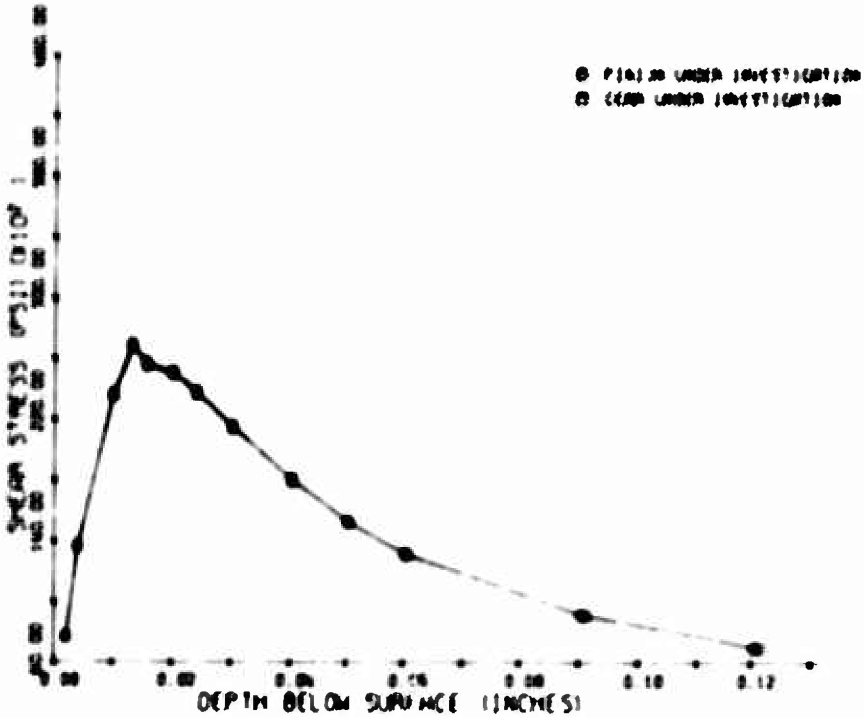

Figure 25. Shear Stress Diagram, Twin 500 HP, 1200-Hour TBO, Ring-Planet Mesh.

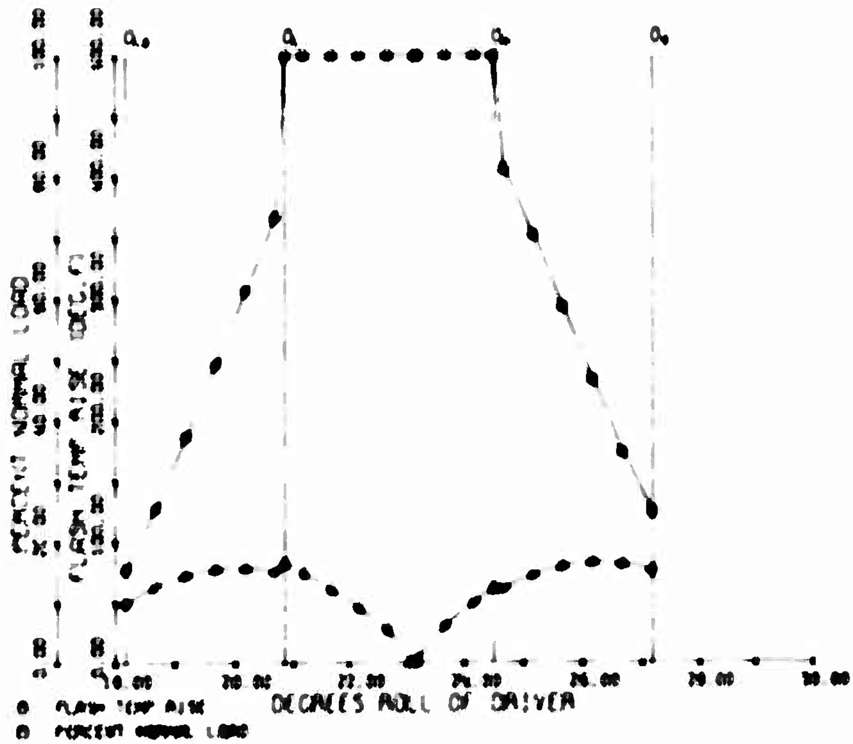

Figure [illegible]6. Flash Temperature Rise and Load Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

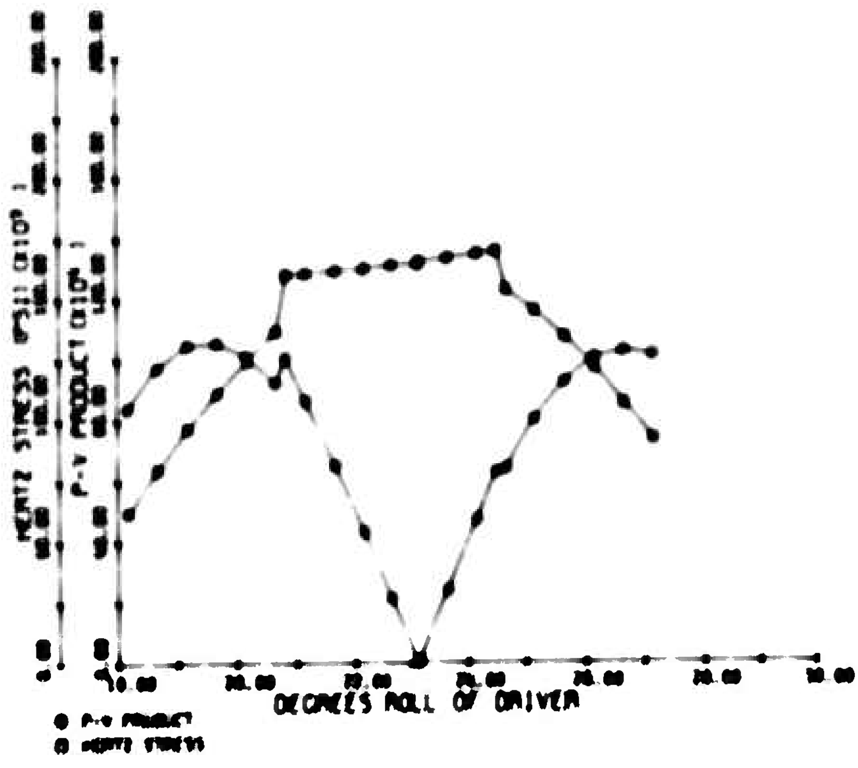

Figure [illegible]. Hertz Stress and P-V Product Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

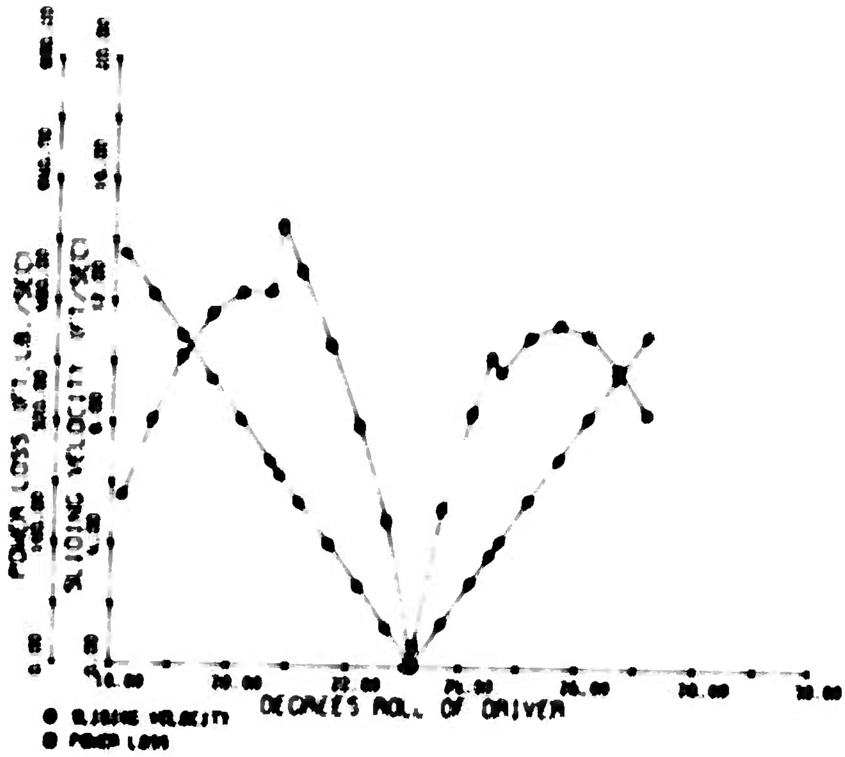

Figure 78. Sliding Velocity and Power Loss Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

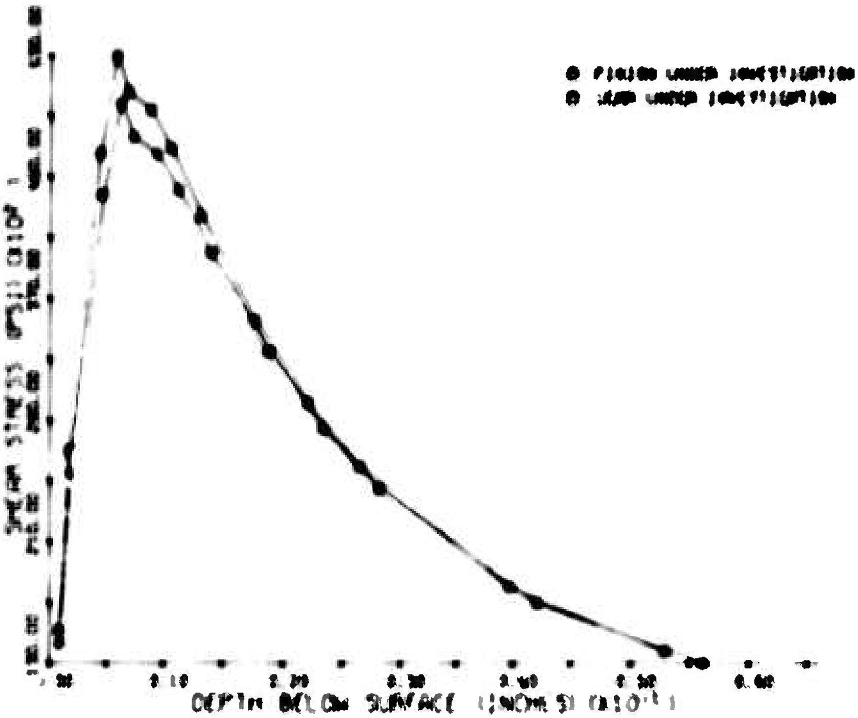

Figure 29. Shear Stress Diagram, Twin 1500 HP,
1200-Hour TBO, Lower Sun-Planet Mesh.

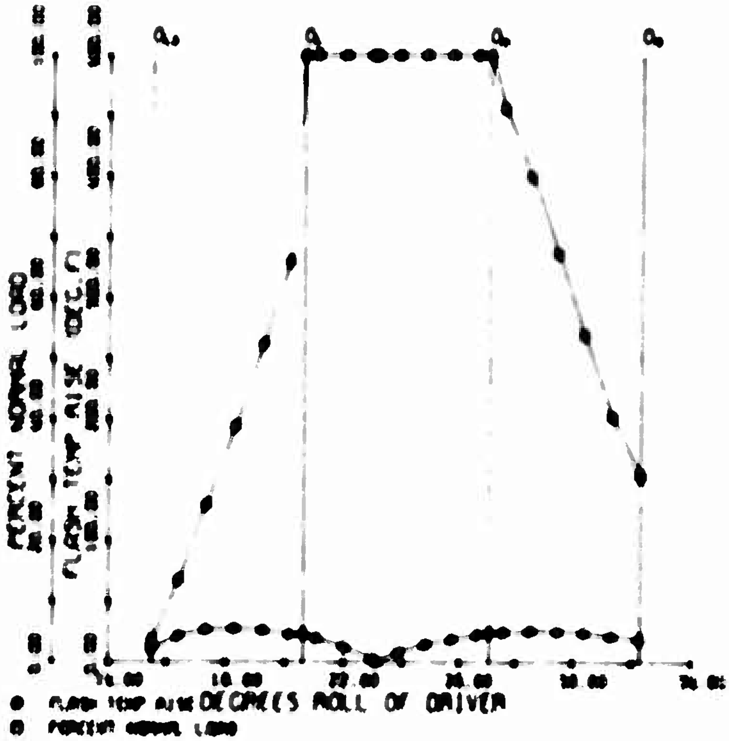

Figure 30. Flash Temperature Rise and Load Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Ring-Planet Mesh.

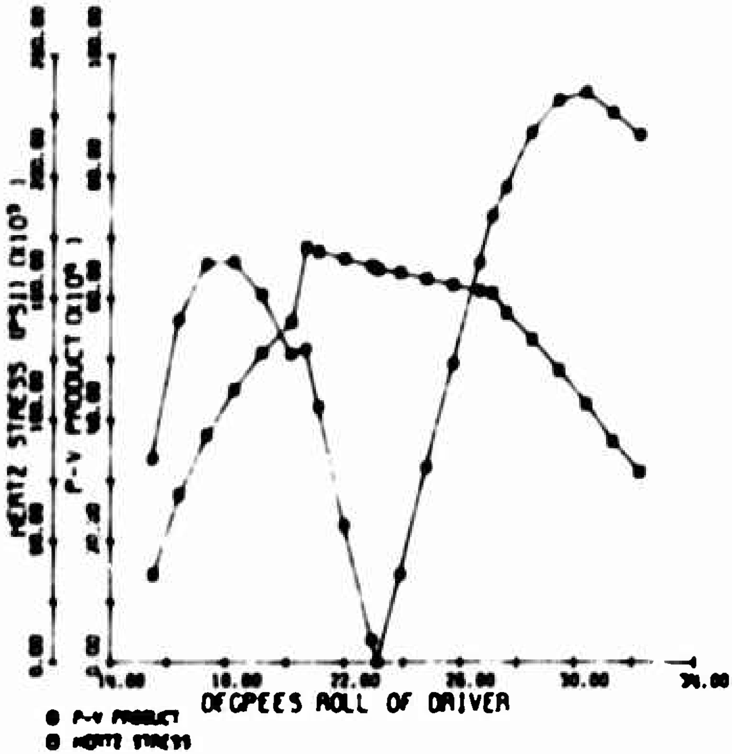

Figure 31. Hertz Stress and P-V Product Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Ring-Planet Mesh.

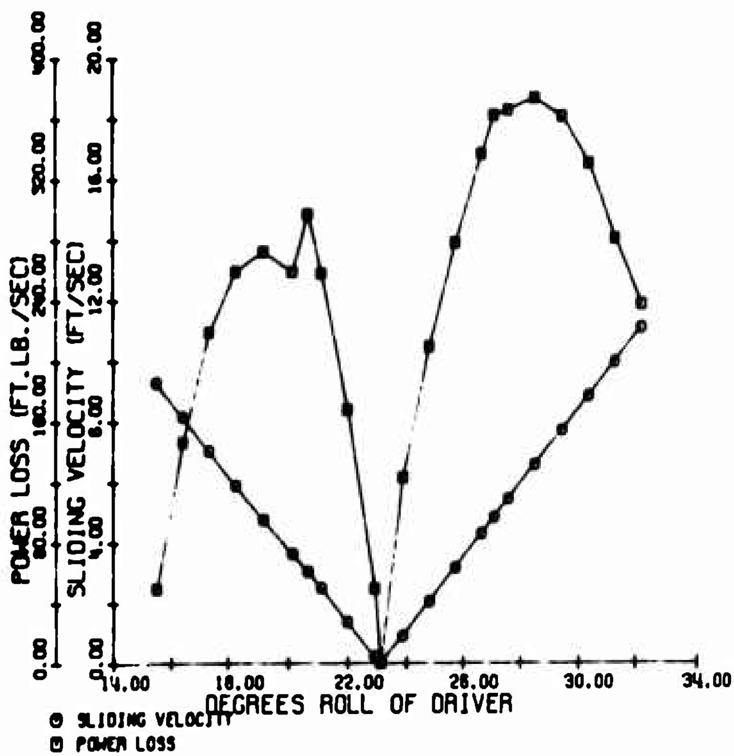

Figure 32. Sliding Velocity and Power Loss Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Ring-Planet Mesh.

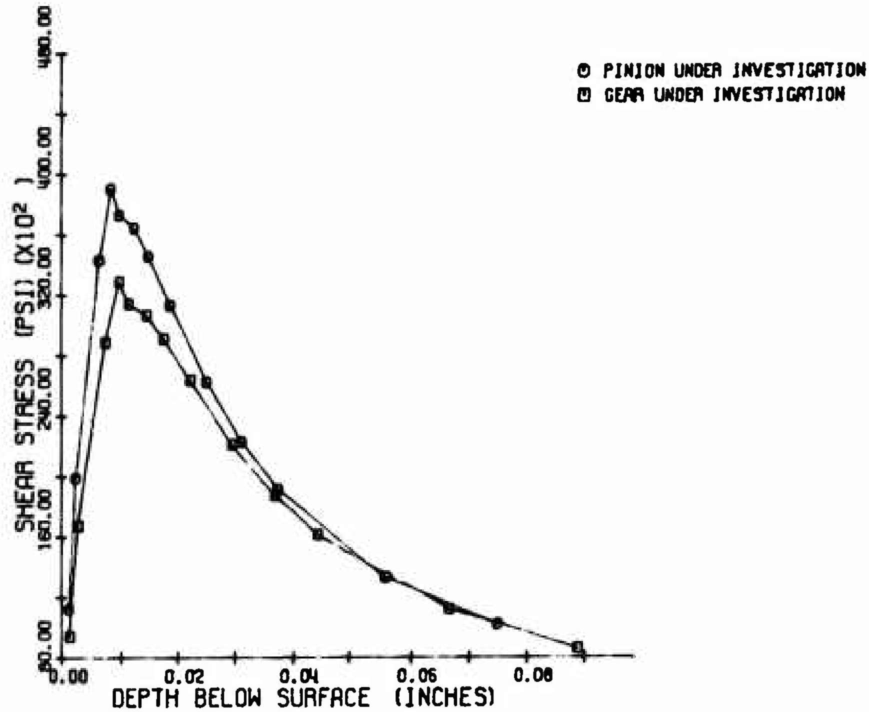

Figure 33. Shear Stress Diagram, Twin 1500 HP, 1200-Hour TBO, Lower Ring-Planet Mesh.

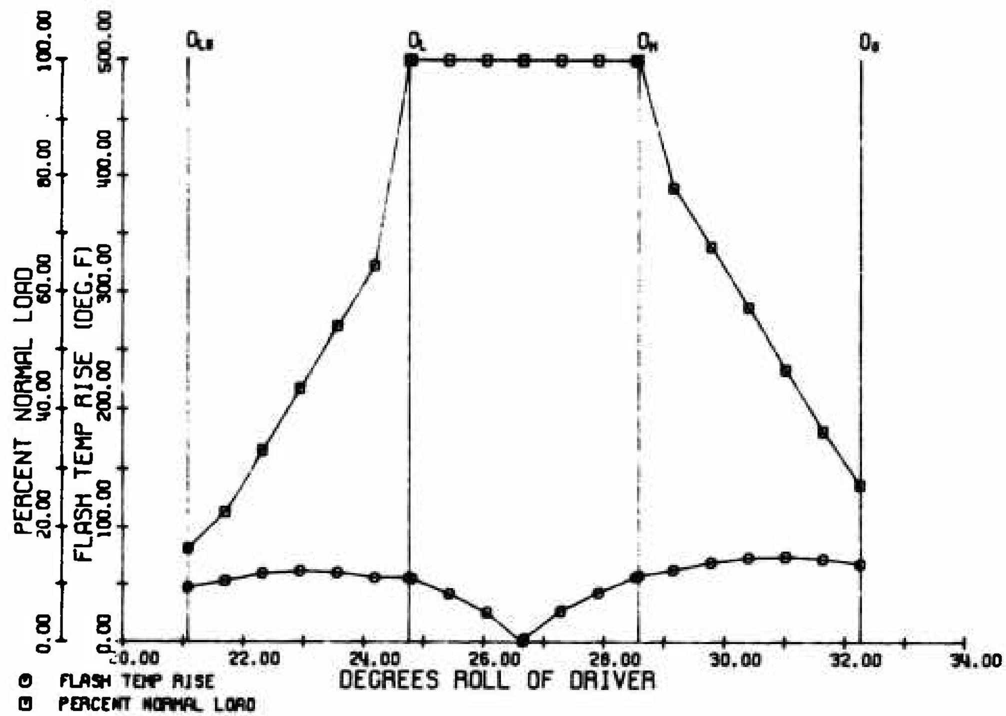

Figure 34. Flash Temperature Rise and Load Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

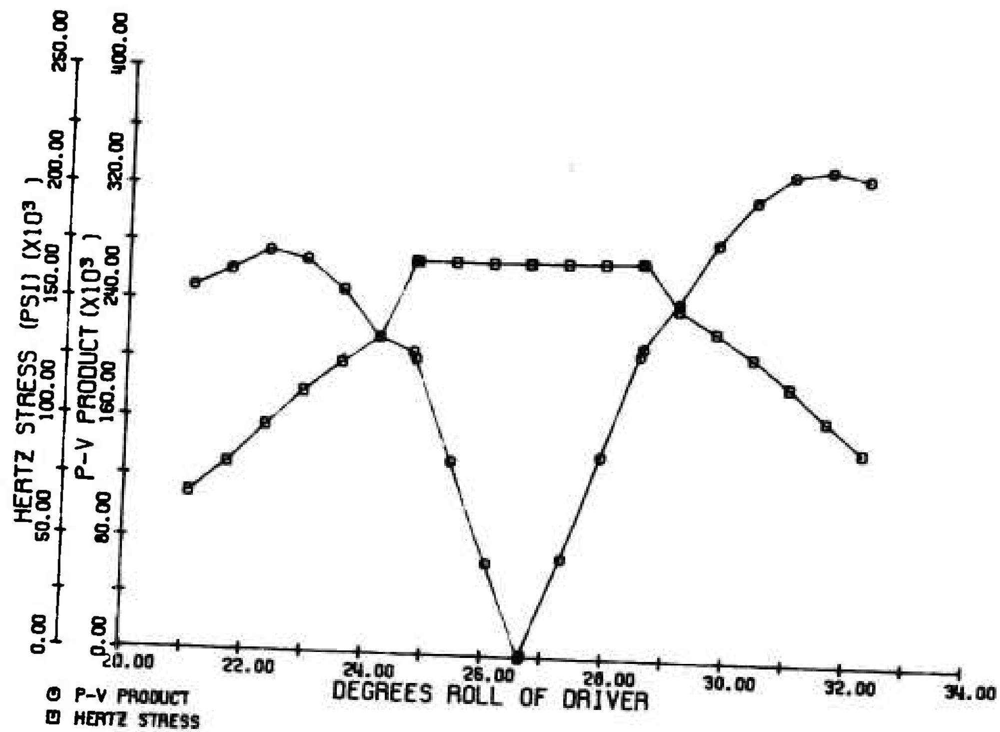

Figure 35. Hertz Stress and P-V Product Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

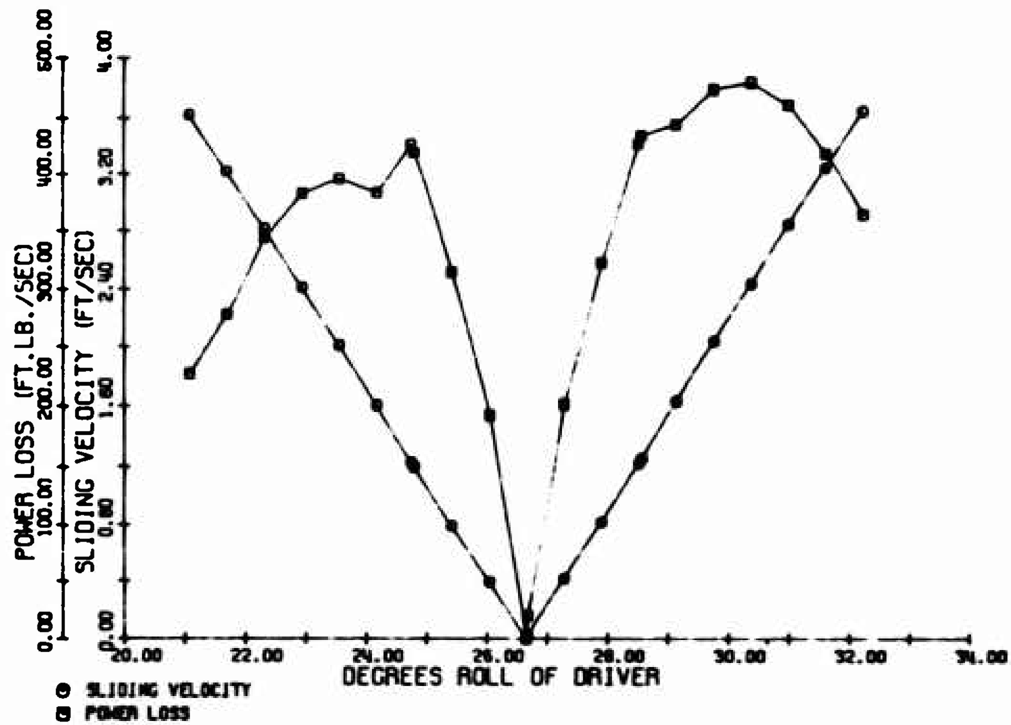

Figure 36. Sliding Velocity and Power Loss Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

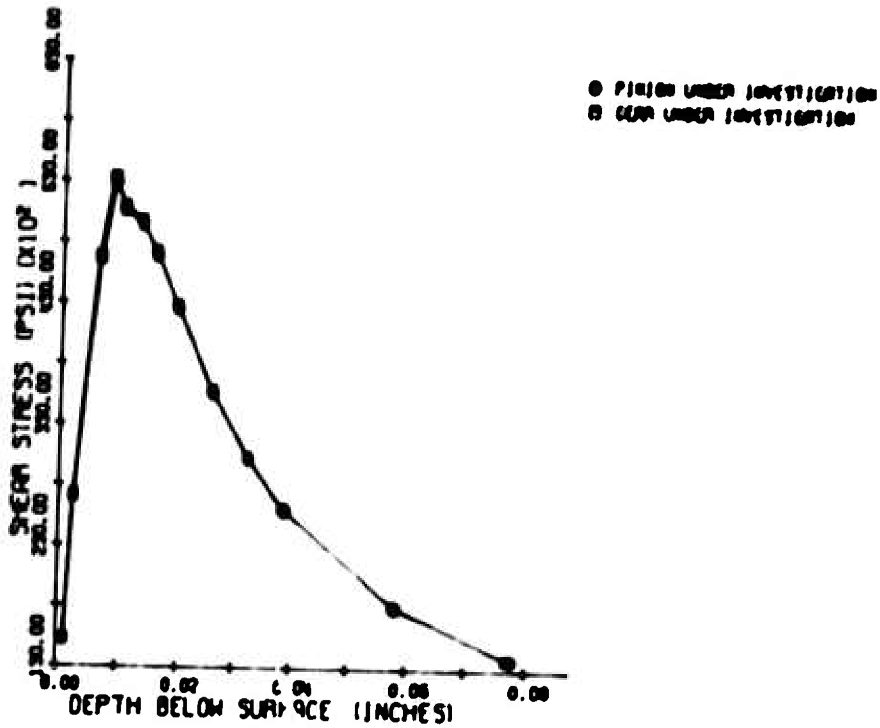

Figure 37. Shear Stress Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

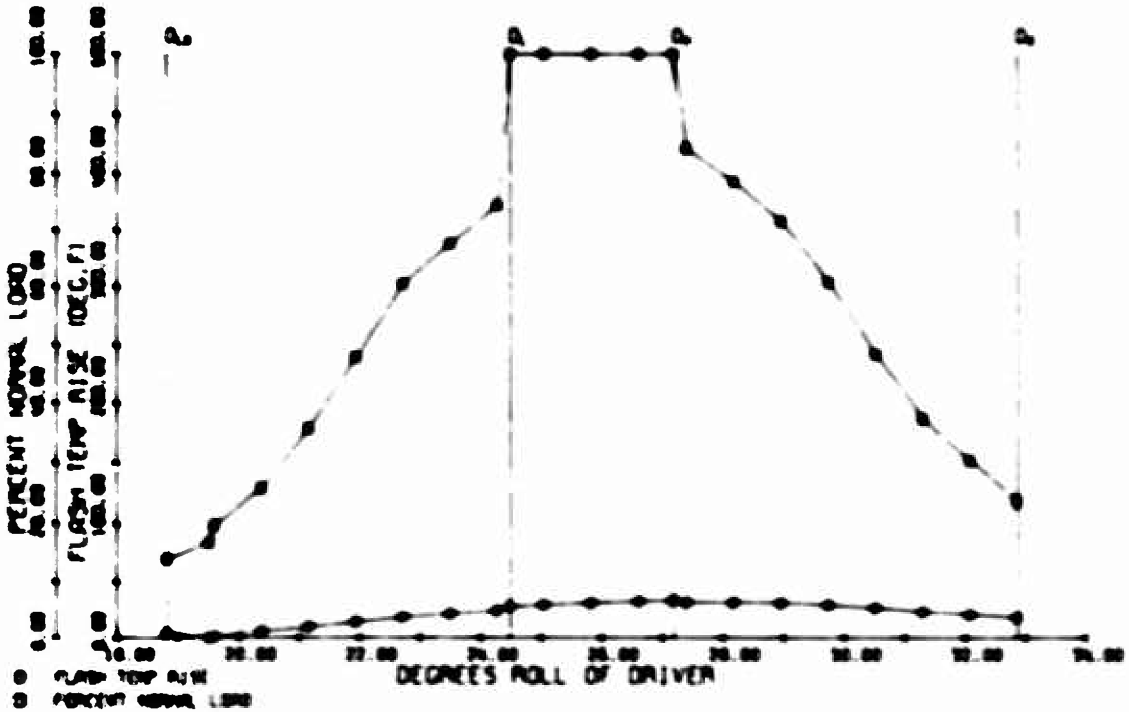

Figure 38. Flash Temperature Rise and Load Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Ring-Planet Mesh.

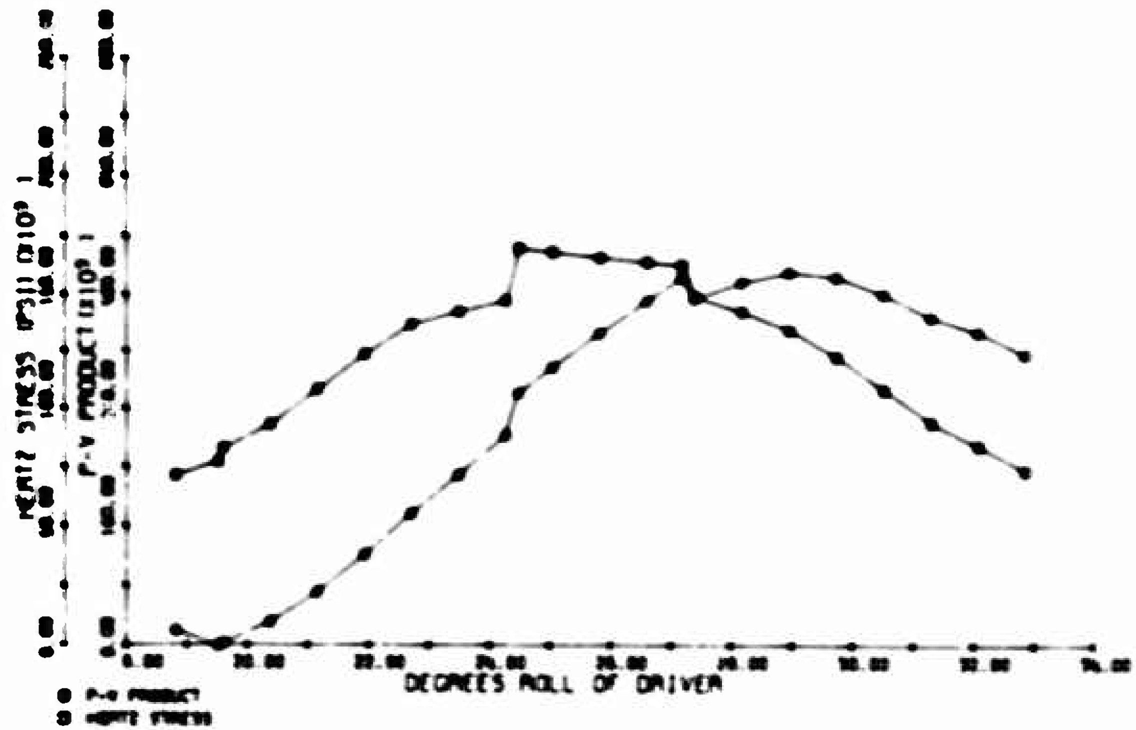

Figure 39. Hertz Stress and P-V Product Diagram. Twin 1300 HP, 1200-Hour TBO, Upper Ring-Planet Mesh.

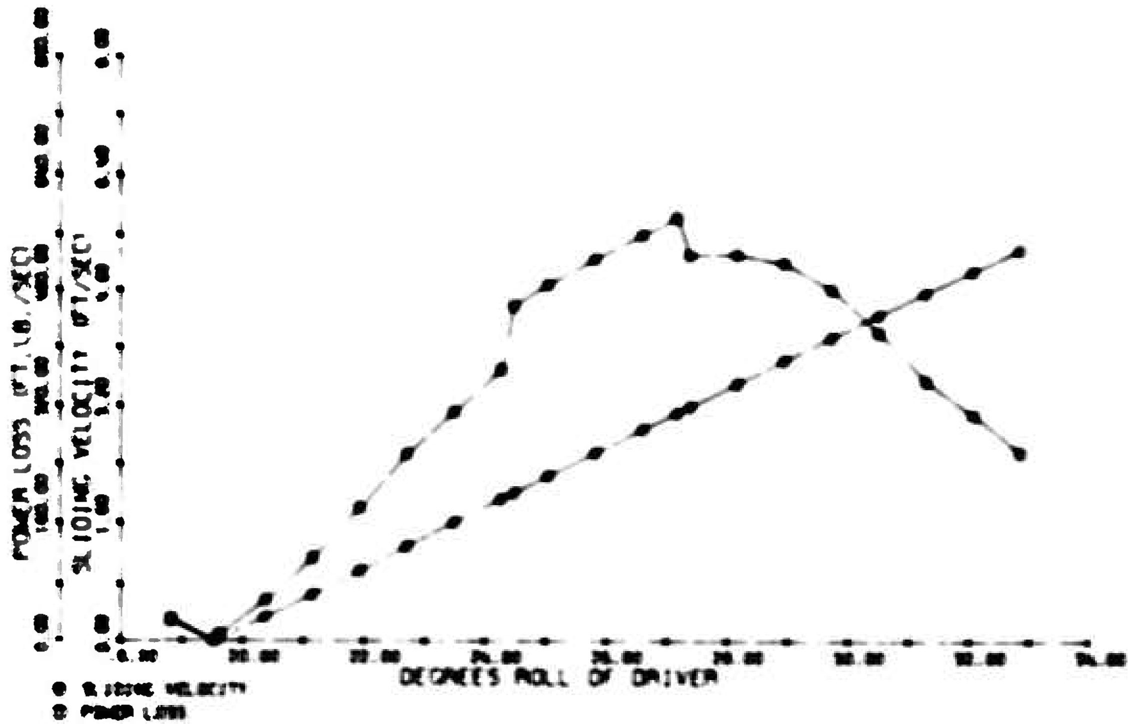

Figure 40. Sliding Velocity and Power Loss Diagram, Twin 1500 HP, 1200-Hour TBO, Upper Ring-Planet Mesh.

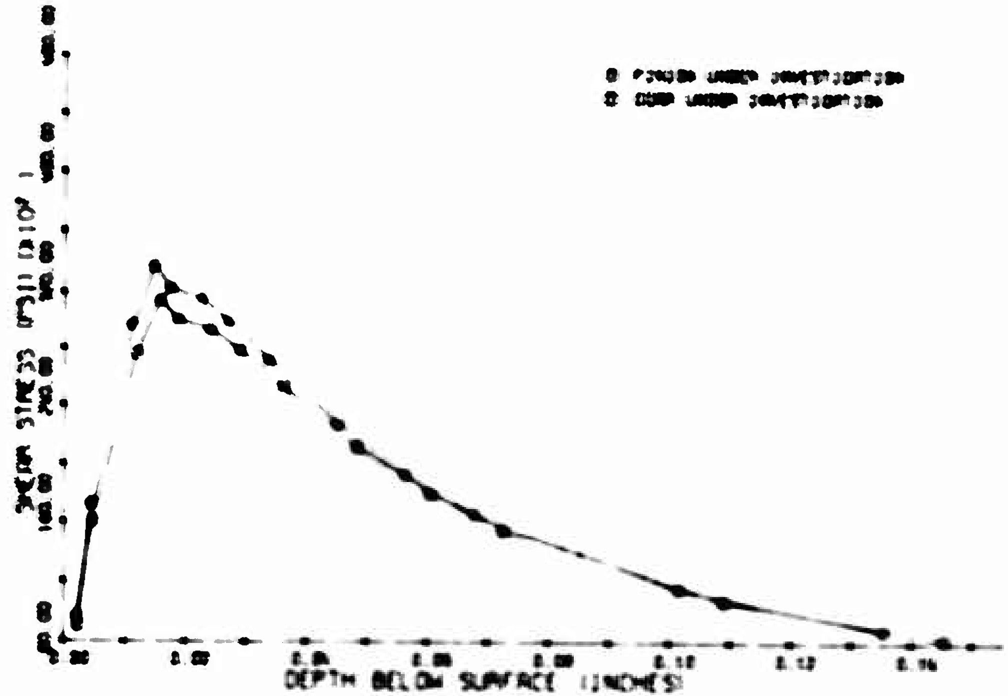

Figure 41. Shear Stress Diagram, Twin 1500 HP,
1200-Hour TBO, Upper Ring Planet Mesh.

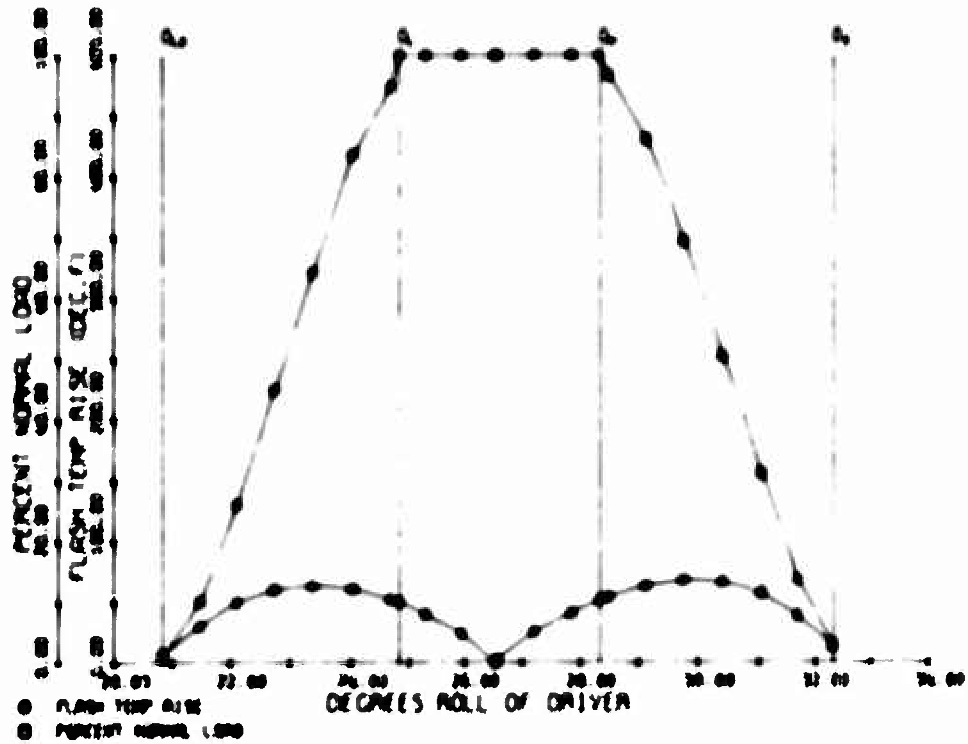

Figure 42. Flash Temperature Rise and Load Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

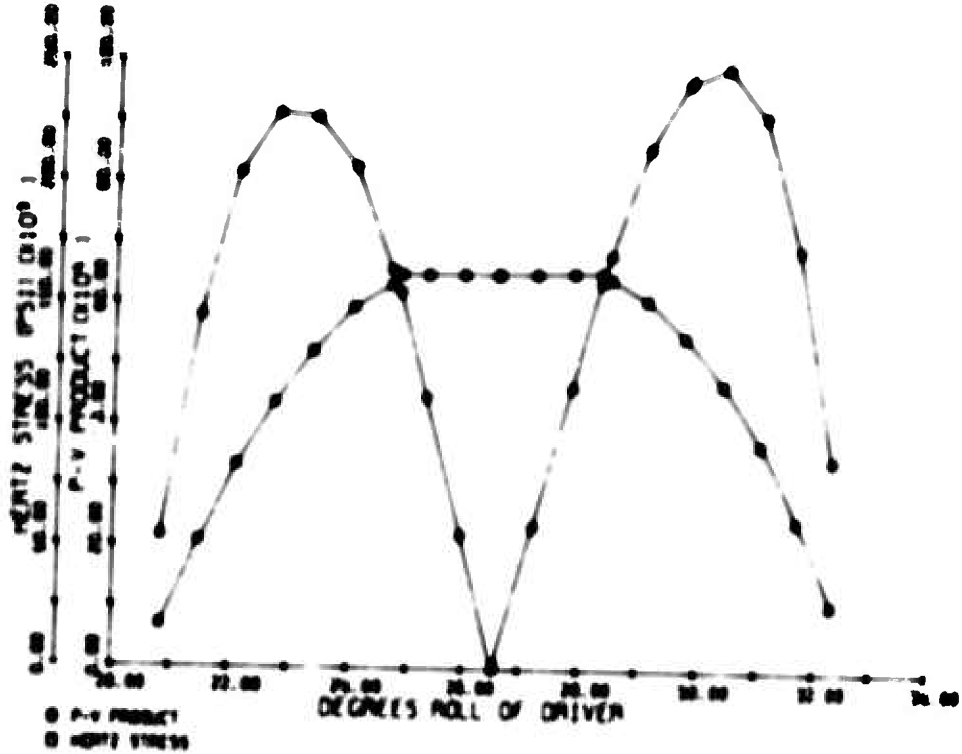

Figure 43 Hertz Stress and P-Y Product Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

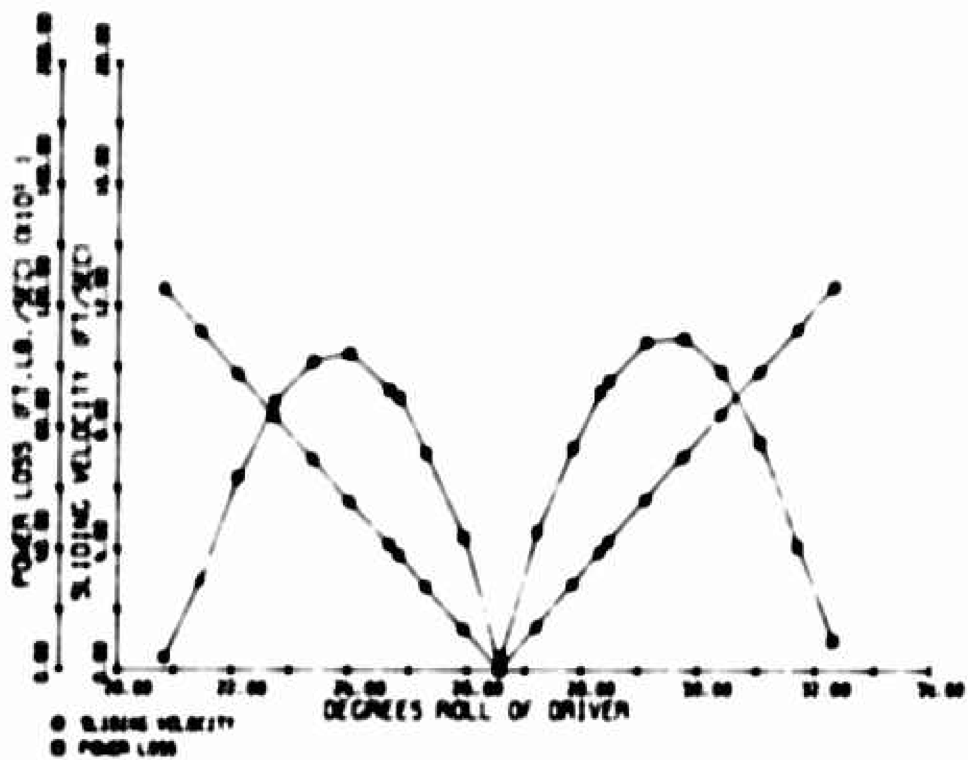

Figure 44   Sliding Velocity and Power Loss Diagram, Twin 4600 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

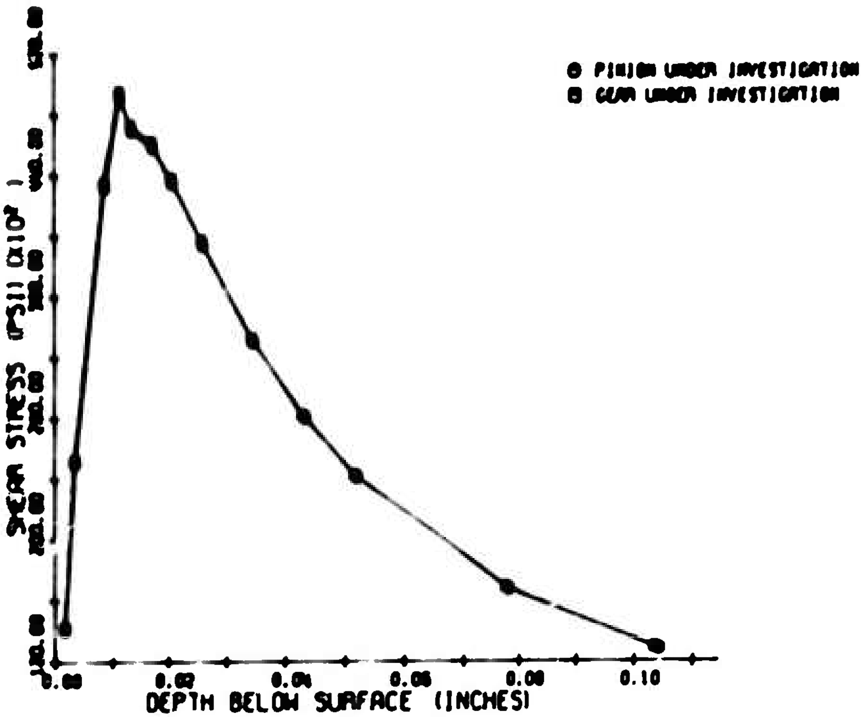

Figure 45. Shear Stress Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Sun-Planet Mesh.

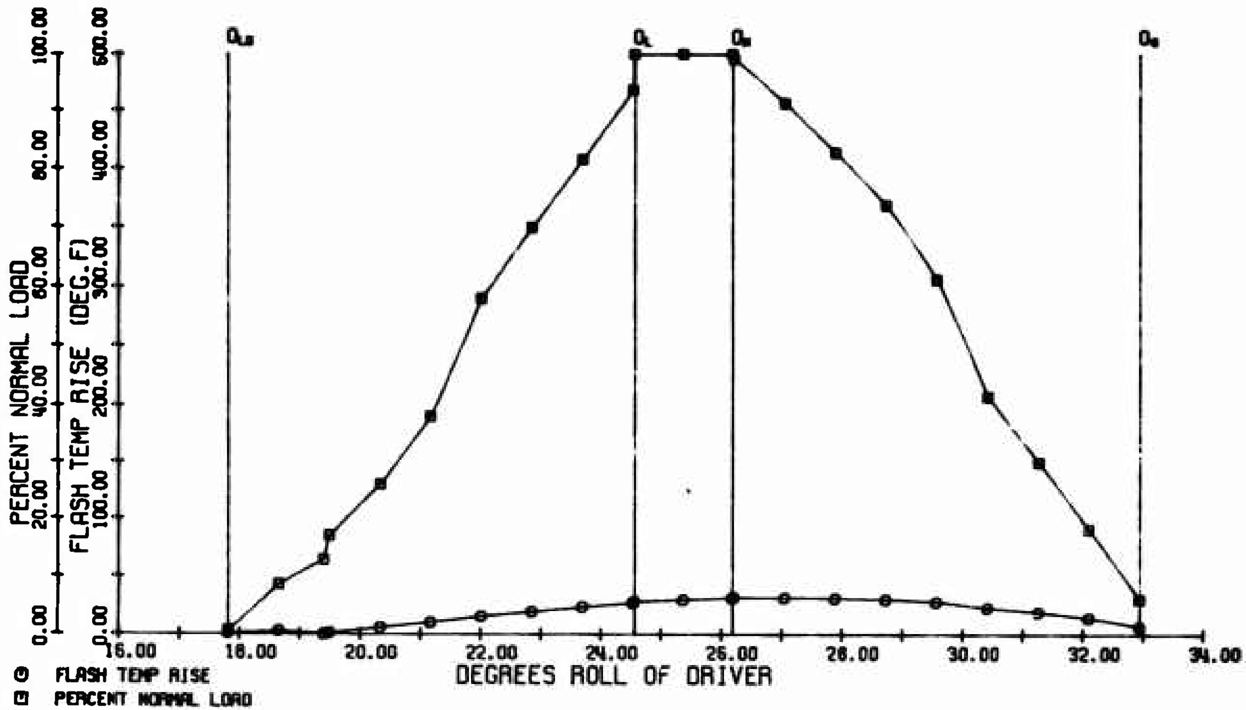

Figure 46. Flash Temperature Rise and Load Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Planet-Ring Mesh.

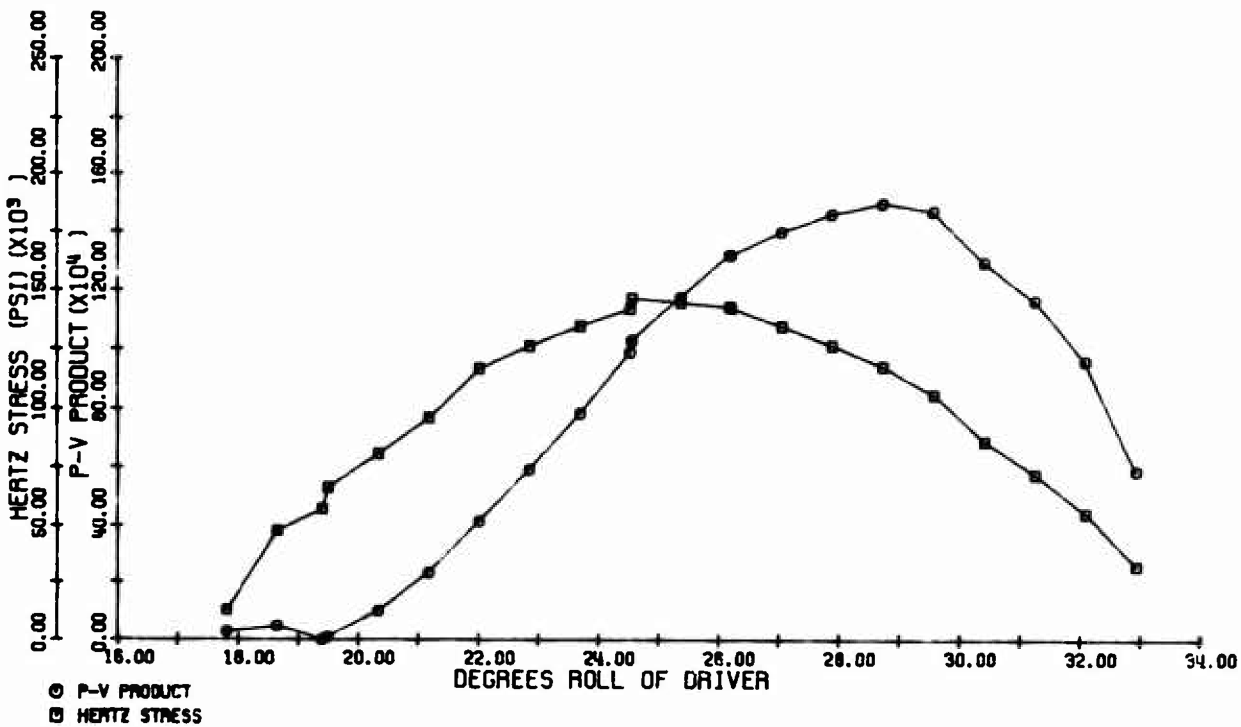

Figure 47. Hertz Stress and P-V Product Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Planet-Ring Mesh.

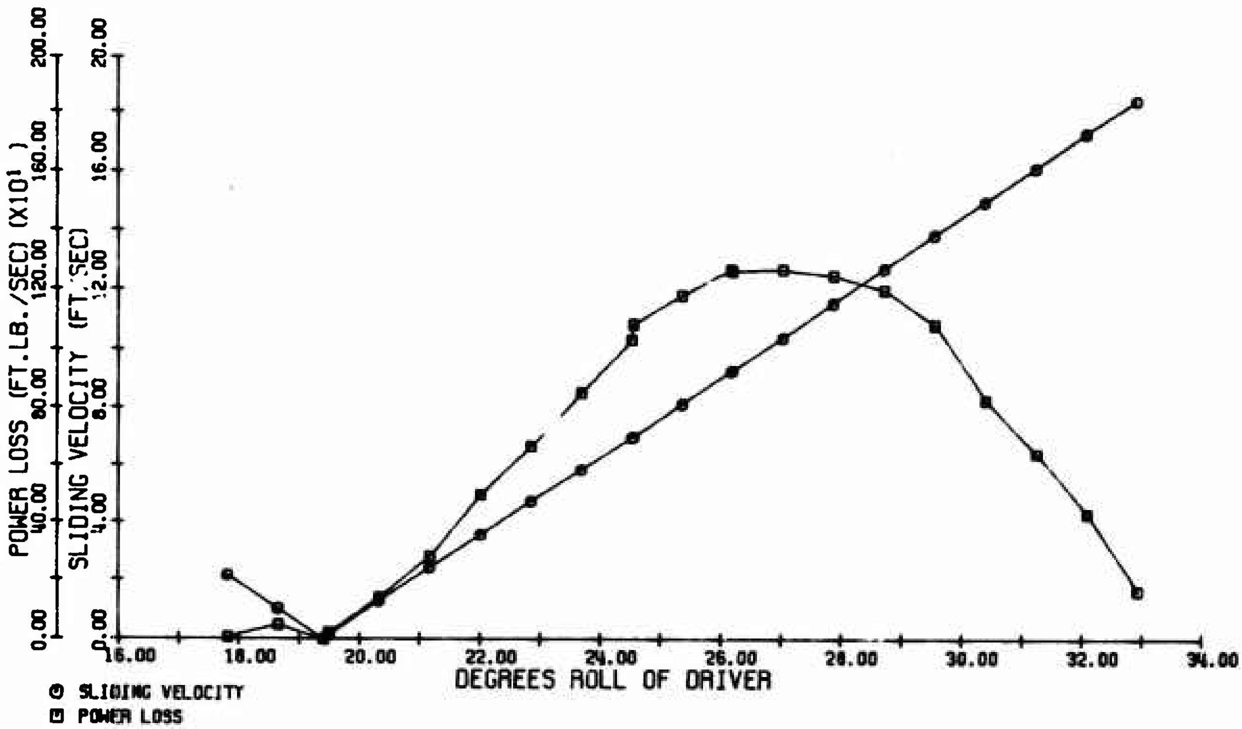

Figure 48. Sliding Velocity and Power Loss Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Planet-Ring Mesh.

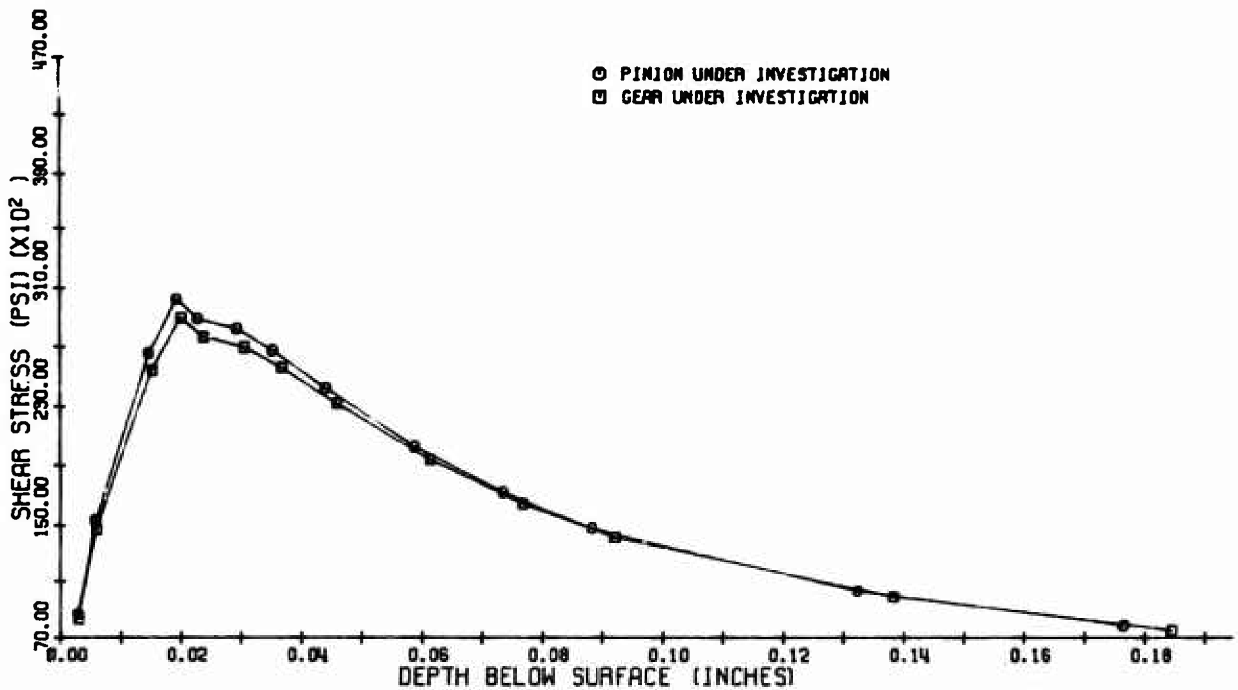

Figure 49. Shear Stress Diagram, Twin 4800 HP, 1200-Hour TBO, Lower Planet-Ring Mesh.

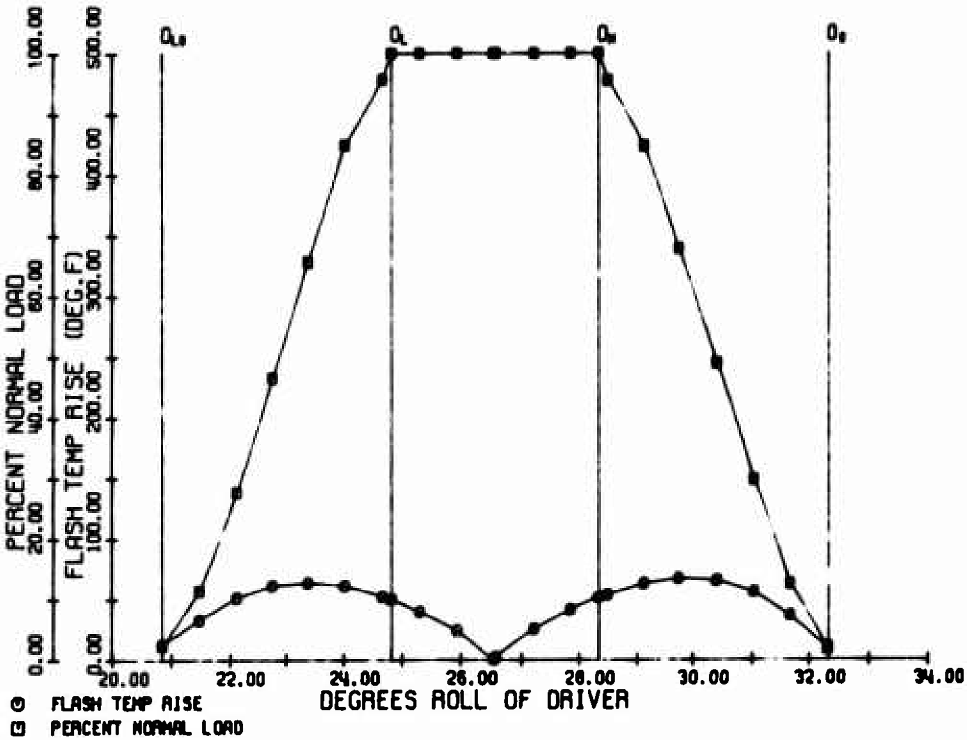

Figure 50. Flash Temperature Rise and Load Diagram, Twin 4800 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

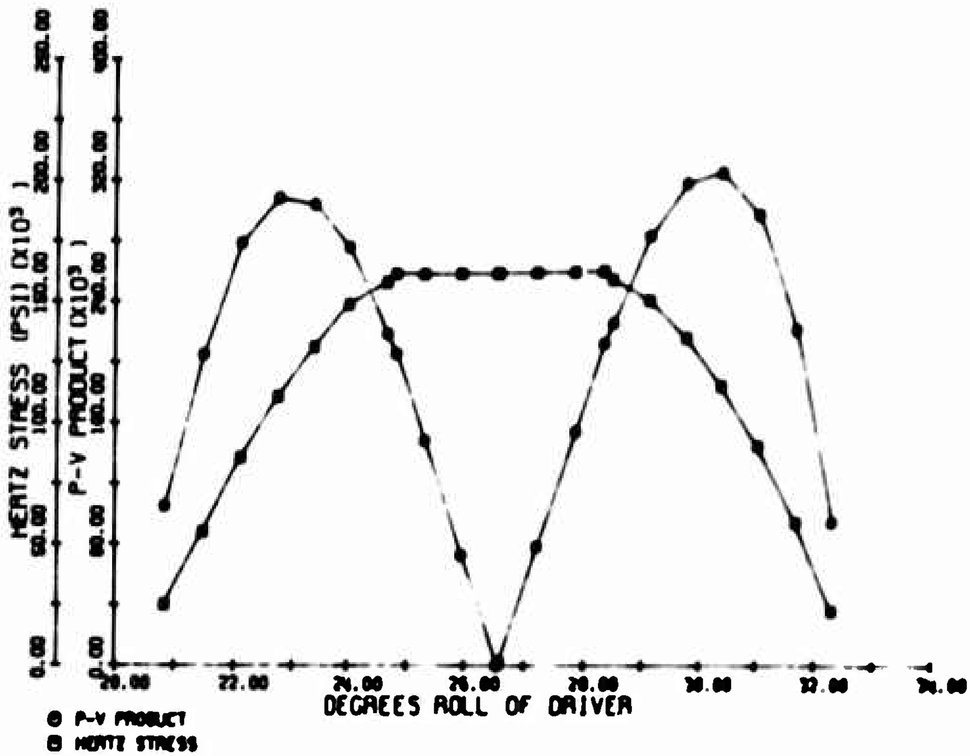

Figure 51. Hertz Stress and P-V Product Diagram, Twin 4800 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

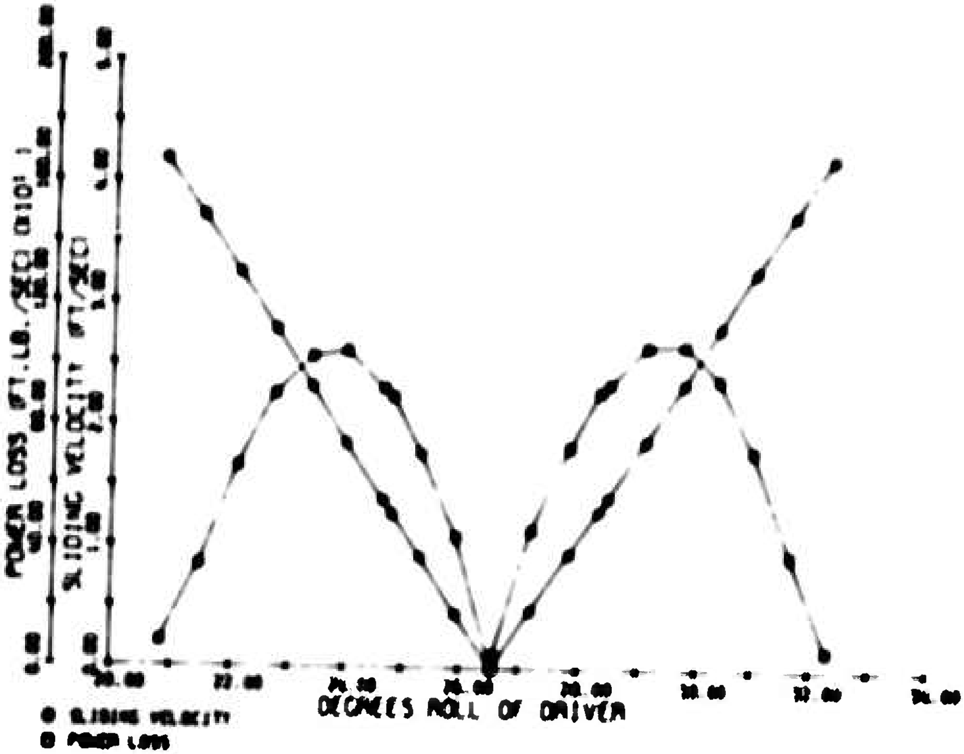

Figure 52. Sliding Velocity and Power Loss Diagram, Twin 4600 HP, 1200 Hour TBO, Upper Sun-Planet Mesh.

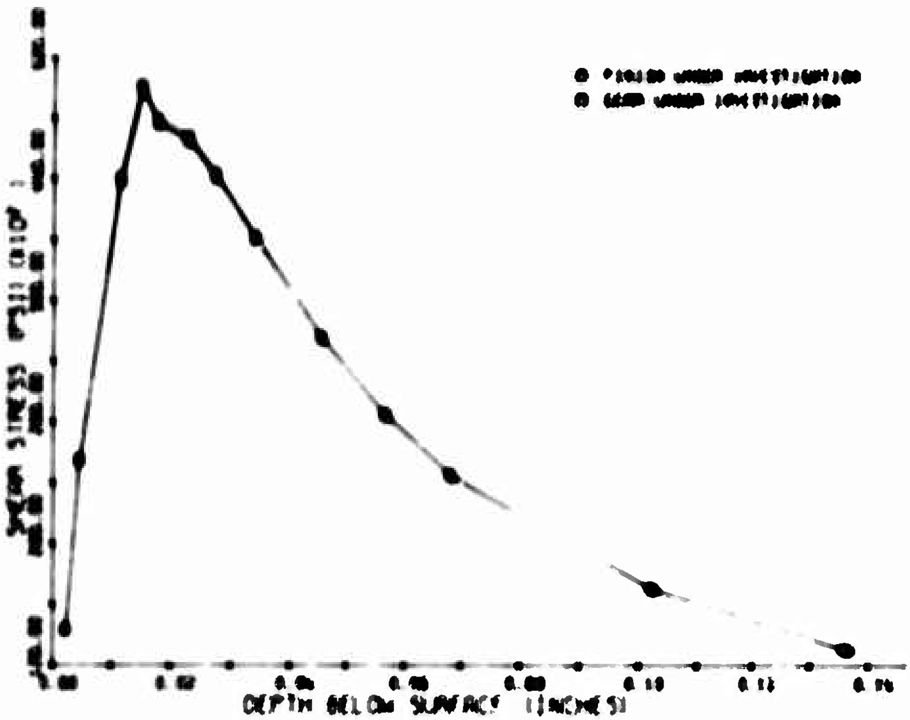

Figure 53. Shear Stress Diagram, Twin 4000 HP, 1200-Hour TBO, Upper Sun-Planet Mesh.

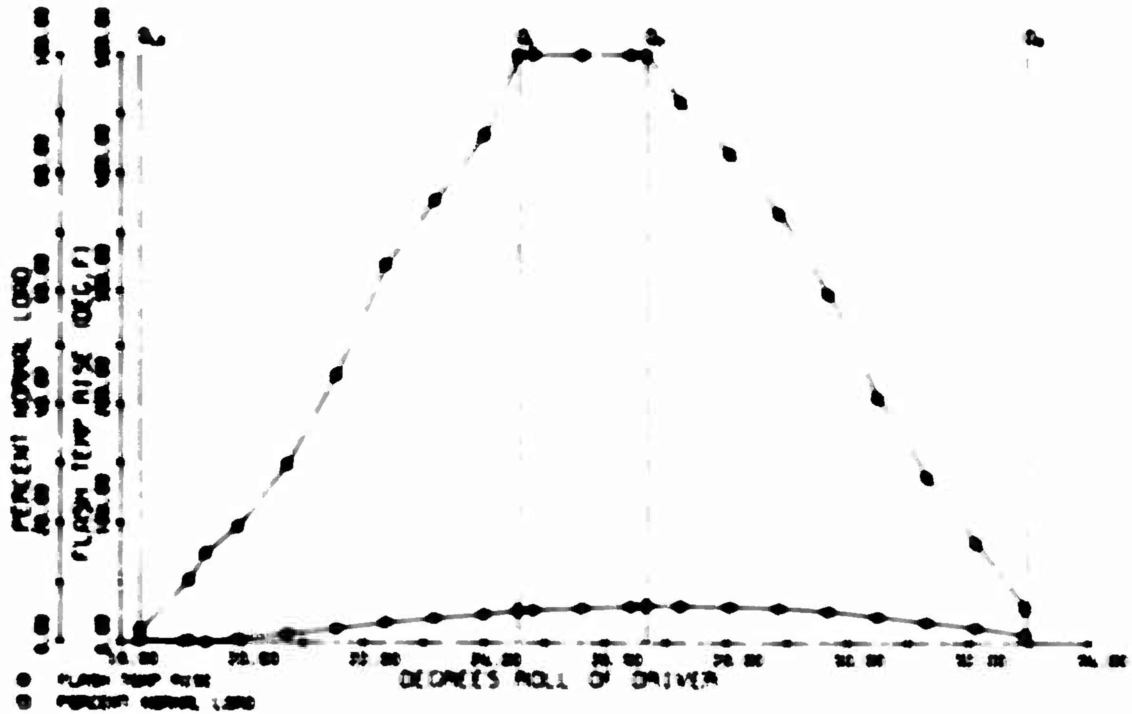

Figure 34. Flash Temperature Rise and Load Diagram. Twin 4600 HP, 1200-Hour TBO, Upper Planet-Ring Mesh.

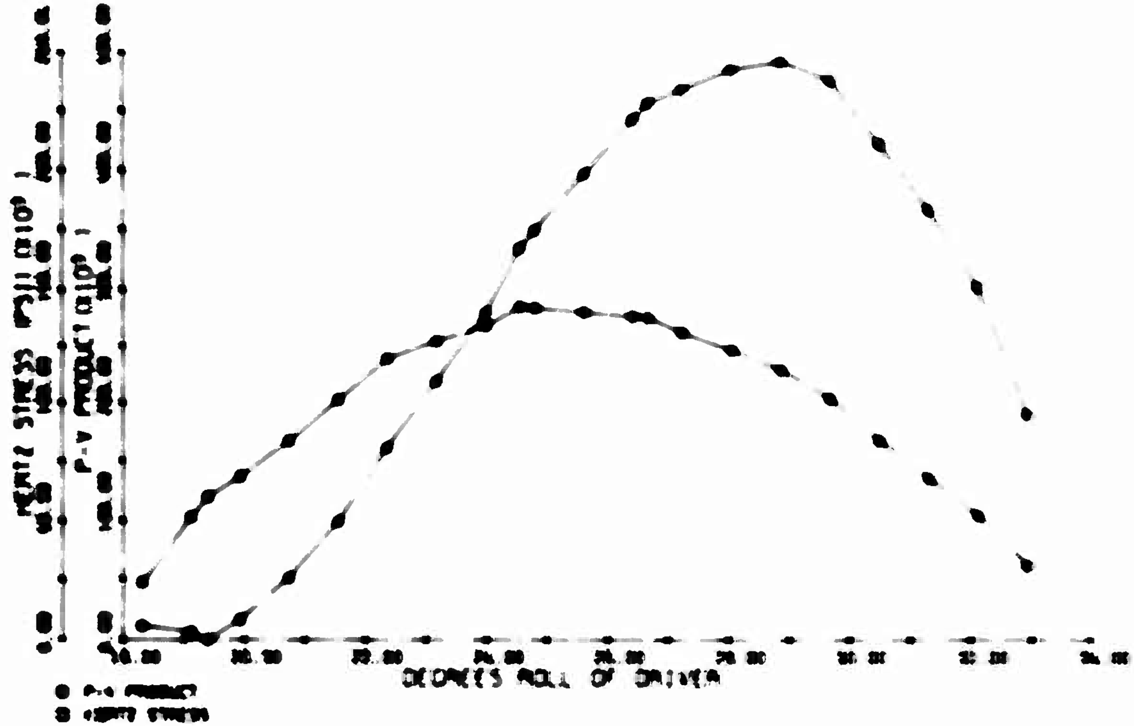

Figure 55. Hertz Stress and P-V Product Diagram, Twin 4800 HP, 1200-Hour TBO, Upper-Planet Ring Mesh.

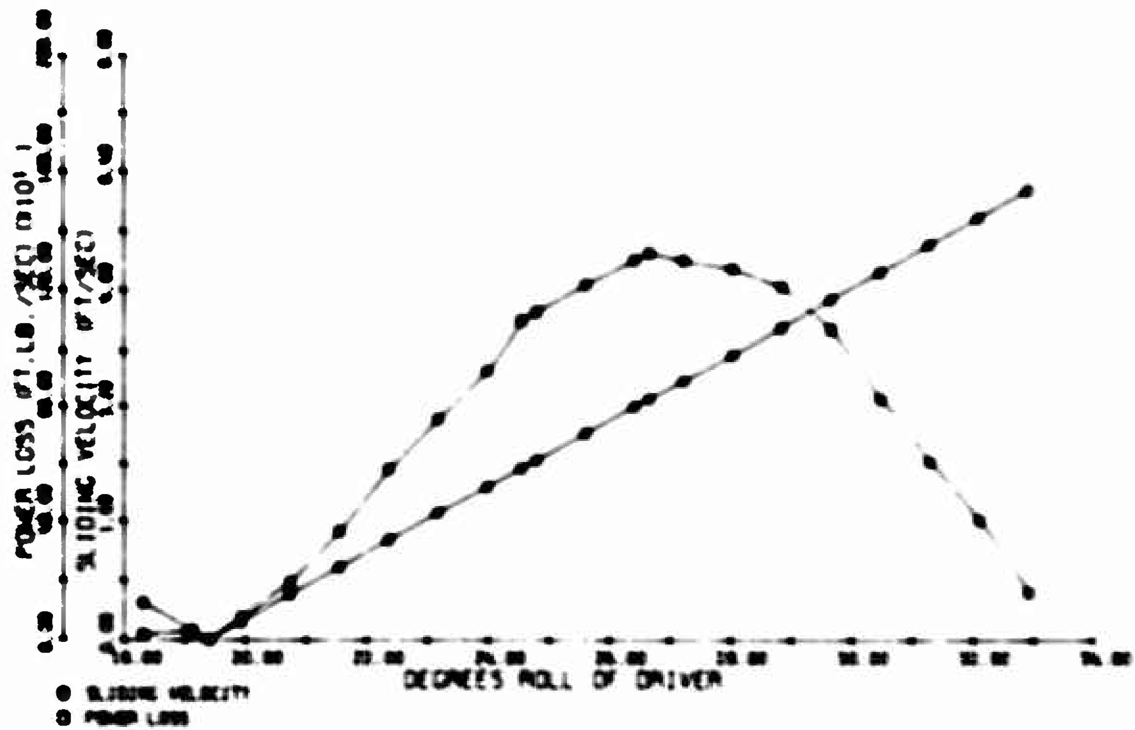

Figure 56. Sliding Velocity and Power Loss Diagram. Twin 4800 HP, 1200-Hour TBO, Upper Planet-Ring Mesh.

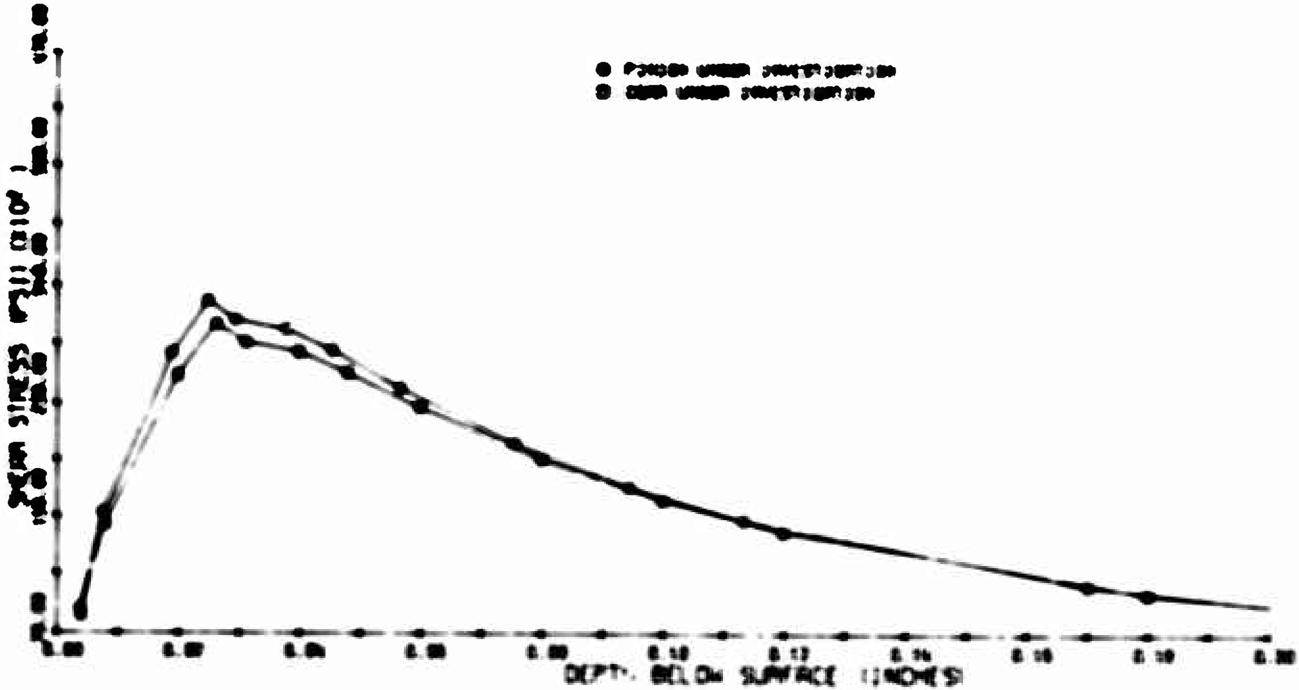

Figure 57. Shear Stress Diagram, Twin 4800 HP, 1200-Hour TBO, Upper Planet-Ring Mesh.